SHAFT SEALS FOR DYNAMIC APPLICATIONS

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Preface

All machines must be sealed to retain lubricants and other fluids. Seals also prevent the entry of contaminants, which can destroy bearings, gears, and other moving components contained within the machine housing. Seals can be classified into two major categories: dynamic and static. Radial lip shaft seals are the most widely used seal types throughout industry today for rotating, oscillating, and reciprocating shaft applications.

Shaft Seals for Dynamic Applications> is a practical comprehensive reference handbook designed for seal users, seal designers, lubrication engineers, maintenance personnel, and chemists, and it can also be used as a teaching guide for seminars.

Chapter 1 is an introduction that describes all the seal types used in industry today and classifies them into dynamic and static applications. The radial lip seal is defined and the importance of this seal type for dynamic applications is stressed.

The heart of the radial lip seal is the elastomer that is molded into the final lip shape. Elastomers are compounded from synthetic oil-resistant polymers and other important fillers. Chapter 2 describes the polymers used to make synthetic elastomers, outlines test procedures for unvulcanized and vulcanized elastomeric materials, and explains the standard ASTM classification system for elastomeric materials.

One of the important functions of the radial lip shaft seal is to prevent lubricants from leaking out of the machine element. A comprehensive review of lubricants is found in Chapter 3, which describes the base oils and additives used to make a lubricating oil. The properties and test procedures used to classify lubricating greases, engine oils, gear oils, automatic transmission fluids, and specialty lubricants are described in clear terms. Test procedures for evaluating seal materials in the various lubricants are outlined. Interaction of elastomers with lubricants is explained with comprehensive test data based on my experience.

Many radial lip shaft seal designs exist to meet the needs of specific applications. These designs are discussed in detail and the reasons for the design variations are explained in detail in Chapter 4.

The Society of Automotive Engineers (SAE), the Rubber Manufacturers Association (RMA), the American Society of Testing and Materials (ASTM), the American Society of Tribology and Lubrication Engineers (ASTL), and other organizations have developed recommended practices for radial shaft seal selection, testing specifications and installation. Chapter 5 summarizes these recommended practices for an easy review.

The principles of seal operation are discussed in Chapter 6, which explores the theories describing the sealing mechanism and presents the most probable explanation of seal function. Test procedures used to measure seal operating characteristics and seal life appear in Chapter 7.

In Chapter 8, practical seal test data are given to explain how variations in an application's operating conditions (sump temperature, shaft speed, shaft eccentricity, seal cocking, lubricants, lubricant fill level, pressure, external contaminants) affect seal performance and life.

Practical seal test data are used to explain how variations in seal design parameters will affect seal performance and life in Chapter 9. The effects of garter spring tension, "R" value, and spring interference are discussed in detail. Changes in seal design affect performance in low temperature, pressure, and dusty applications. Data are presented and the most effective designs to give the optimum performance under these conditions are recommended.

Chapter 10 describes the sealing system and provides procedures to determine the cause of field leakage problems. Typical practical sealing applications are examined, and the choice of seal design and material for each of these different applications is explained in Chapter 11.

The understanding of performance and design needs of any product is enhanced if the manufacturing methods used to make that product are understood. The manufacturing methods used to produce radial lip shaft seals are described in Chapter 12. Seal users and designers will be able to specify and design seals that function in a particular application without compromising the manufacturability of the seal.

Many test procedures exist to ensure the quality of radial lip shaft seals. These procedures are reviewed, explained, and combined in Chapter 13 for easy reference. The

Appendixes include an extensive review of radial lip seal terminology (Appendix A), terminology for elastomers (Appendix B), lubricant terminology (Appendix C), a system of units and eight conversion tables (Appendix D), and a table of nomenclature (Appendix E). An extensive reference list is also included.

This book is a handy, easy reference that will enable the reader to understand all aspects of radial lip shaft seals. When properly used, seal designers, seal users, chemists, and lubricant engineers will be able to work together to select the proper seal, material, and lubricant for dynamic shaft applications used extensively in the transportation, pump, and other machine industries.

Recognition is due to the seal engineers and scientists who have devoted their time to develop the oil seal standards that are found in the RMA documents, the SAE standard books, and the publications of other standards organizations.

Special thanks to Bob Dega, Bob Brink, Jimmy Cooper, Art Ginn, Al Berens, Don Jutzi, Merril Karcher, Jim Symons, Stan Smith, Frank Charhut, John Serio, and Dennis Otto for pioneering RMA and SAE committees that advanced oil seal science. Their work has helped to make this book possible. I am also deeply indebted to Dan Czernik, David South, and Patty Rose for their help and encouragement.

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Les Horve

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1

Introduction

1.1 SEAL TYPES

Modern society is highly dependent upon machinery to perform numerous essential tasks that range from transportation to household chores. These machines require many different types of seals to retain lubricants, liquids, solids, and gases [19, 23, 24, 76, 87, 187, 221, 130a, 233]. Special seal designs are required to contain pressure and vacuum. Sealing devices also prevent foreign particles from entering the sealed cavity, as they can damage machinery components and lead to premature failure. The application conditions can vary considerably, and many seal designs have evolved to satisfy these conditions.

Some applications will tolerate a small amount of leakage, whereas others cannot allow any leakage. In theory, zero leakage does not exist, and attempts to eliminate leakage entirely may result in higher seal costs than need be if slight leakage can be tolerated. The higher costs may be warranted if the sealed fluid can damage the environment or cause damage to expensive machinery if lubrication levels become too low. Small amounts of leakage will result in big losses over time (Table 1.1), and the initial seal costs may be of little consequence when evaluating the total cost when fluid leakage exists.

Seals are typically classified as static or dynamic (Fig. 1.1). Static seals are used where there is little or no relative motion between mating sur-

Table 1.1 Fluid Loss

	Fluid loss (gallons)		
Leakage rate	Loss per day	Loss per week	Loss per month
1 drop every 10 sec	0.1	0.7	3.0
1 drop every 5 sec	0.2	1.4	6.0
1 drop per sec	1.00	7.0	30.0
3 drops per sec	3.00	21.0	90.0
Drips breaking into a stream	20.0	140.0	600.0

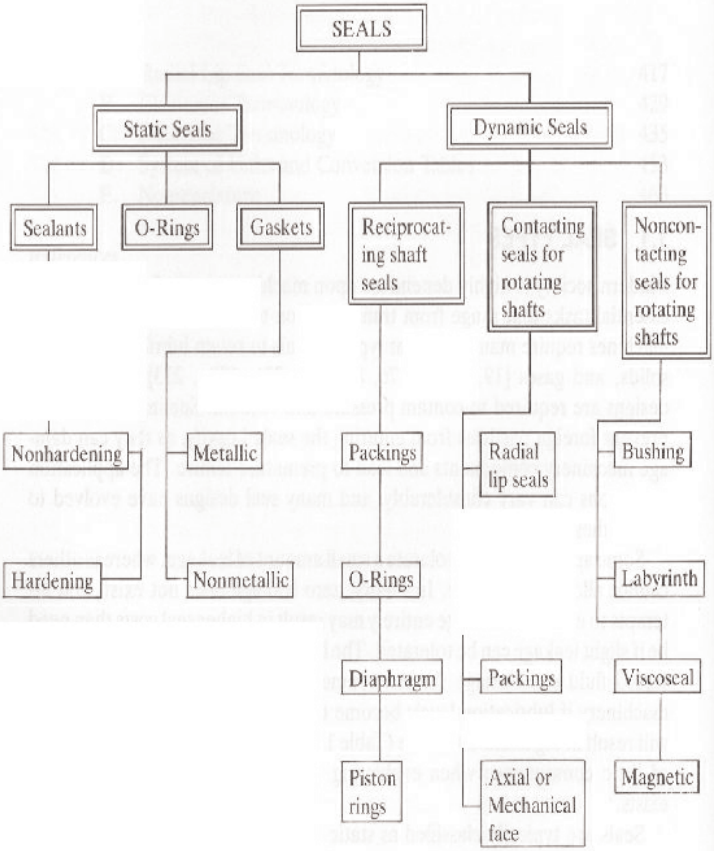
SEALS
Static Seals
Dynamic Seals
Sealants
O-Rings
Gaskets
Reciprocating shaft seals
Contacting seals for rotating shafts
Noncontacting seals for rotating shafts
Nonhardening
Hardening
Metallic
Nonmetallic
Packings
O-Rings
Diaphragm
Piston rings
Radial lip seals
Packings
Axial or Mechanical face
Bushing
Labyrinth
Viscoseal
Magnetic

Fig. 1.1 Seal types. (Courtesy of Chicago Rawhide.)

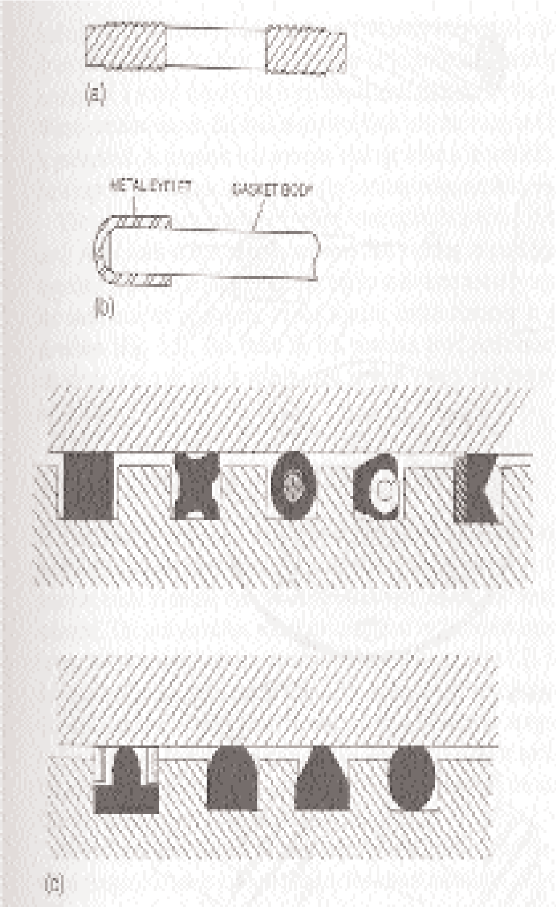

Fig. 1.2 Seal types for static applications. (a) Metal gaskets; (b) nonmetallic gaskets; (c) O-rings and other rubber shapes. (Courtesy of Chicago Rawhide.)

faces. Typical static seals include metal gaskets, nonmetallic or deformable gaskets, elastomeric O-rings and other elastomeric shapes (Fig. 1.2). Liquid sealants are also used to form static seals. Dynamic seals are used to seal shafts that rotate, oscillate, and reciprocate. Packings, O-rings, diaphragms, radial lip seals, and piston rings are used to seal shafts that reciprocate (Fig. 1.3).

Both contacting and noncontacting seal designs can be used to seal shafts that rotate. There are four basic types of seals that function without having direct contact with the shaft. Bushing and labyrinth seals have a clearance between the shaft and the body of the seal. The amount of

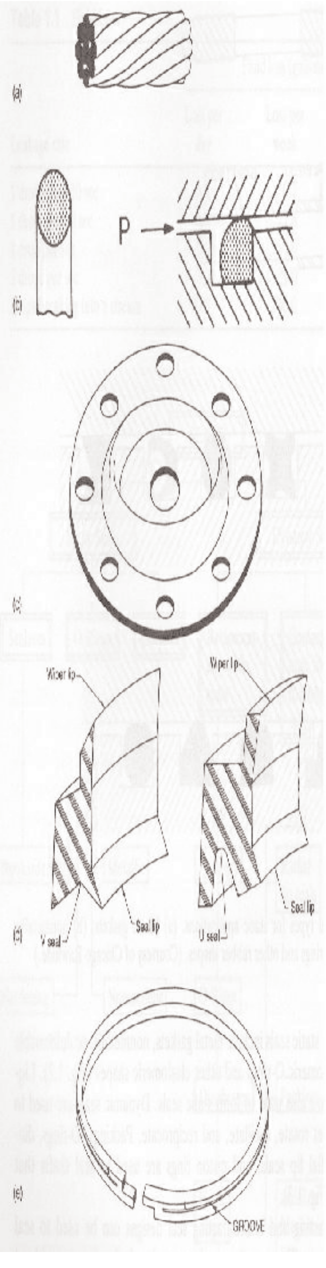
(a)
(b)
P
(c)
Wiper lip
Wiper lip
Seal lip
Seal lip
V seal
U seal
(d)
(e)
GROOVE

Fig. 1.3 Seal types for reciprocating shafts. (a) Packings; (b) O-rings; (c) diaphragms; (d) lip seals; (e) piston rings. (Courtesy of Chicago Rawhide.)

leakage is controlled by changing the seal dimensions to create a pressure drop that throttles the flow. The viscoseal or hydrodynamic seal typically consists of a screw thread that is machined onto the shaft or the housing. When rotation occurs, the fluid is screwed back into the sump and leakage is prevented. A magnetic seal prevents leakage when a ferrofluidic fluid is held captive in the gaps of a labyrinth by a strong magnetic fluid (Fig. 1.4).

The most common method of sealing rotating shafts is with a contacting seal. Axial seals or mechanical face seals use a spring to push two faces together to prevent leakage. Packings and lip seals form a seal by squeezing the shaft and /or by pushing on the housing inside diameter in a radial direction (Fig. 1.5). The radial lip seal provides good performance at a relatively low cost and is widely used in both rotary and reciprocating applications.

1.2 SELECTING A SEAL TYPE FOR AN APPLICATION

It is not easy to determine what type of seal will economically satisfy the needs of a dynamic application. A careful review of the application variables must be made to ensure that the correct seal type, design, and material are selected. The seal variables, which are controlled by the seal designer, may interact with the application variables to create leakage (Table 1.2). Some of the application variables can be controlled by the equipment designer and the end user. It is essential for the seal designer, equipment designer, and end user to work together to obtain the best seal solution for the application. It is suggested that a simple flowchart be followed to eliminate the seal types that will not function in the application (Fig. 1.6).

Many types of seals will function equally well initially, and the user is often tempted to select the seal that has the lowest initial cost. Many times selecting a seal with a low initial cost is the most expensive solution. Inexpensive materials for O-rings, packings, gaskets, etc. may fail early because the application temperature exceeds the material capabilities or because the sealed fluid is aggressive and degrades the elastomer. The low initial price will be quickly offset by expensive repairs, loss of internal fluid, and machine downtime. A higher initial investment in a more expensive material will result in longer life with less downtime and fewer repair bills. A complete analysis of the true cost of a sealing system should include the initial operating costs and other miscellaneous costs that accrue during the life of the equipment.

Initial costs include the purchase price of the seal, special machining operations on the housing and shaft to accommodate the seal, and the time required to assemble, inspect, and test the sealing system integrity.

Operating costs include maintenance, replacement parts, fluid loss, power consumed, and the cost of maintaining an inventory of spare parts.

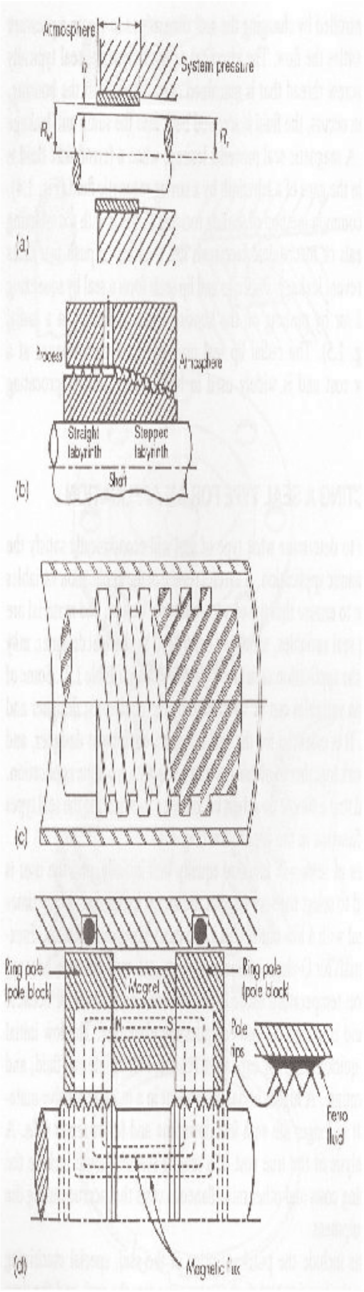
Atmosphere
L
h
System pressure
R_s
R_r
(a)
Process
Atmosphere
Straight labyrinth
Stepped labyrinth
Shaft
(b)
(c)
Ring pole (pole block)
Magnet
Ring pole (pole block)
N
S
Pole tips
Ferro fluid
Magnetic flux
(d)

Fig. 1.4 Noncontacting seal types for rotating shafts. (a) Bushing; (b) labyrinth; (c) visco or hydrodynamic; (d) magnetic. (Courtesy of Chicago Rawhide.)

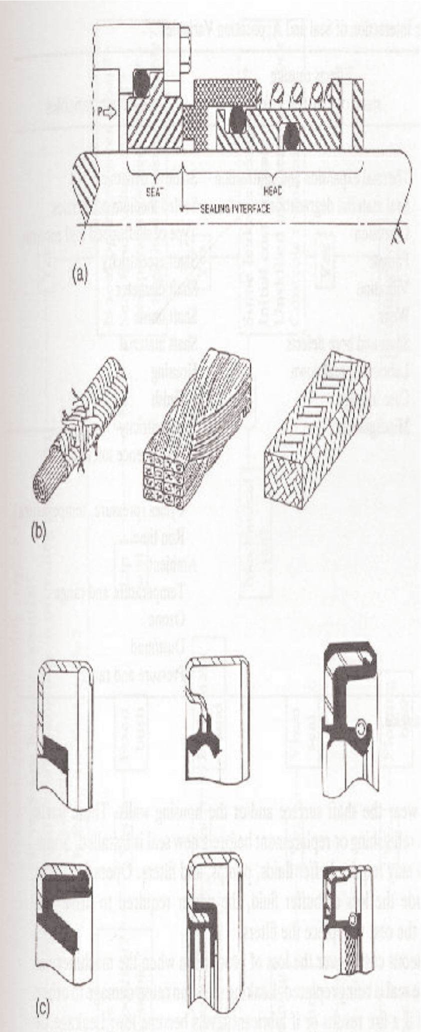
P
SEAT
HEAD
SEALING INTERFACE
(a)
(b)
(c)

Fig. 1.5 Contacting seals for rotating shafts. (a) Axial or mechanical face; (b) packings; (c) lip seals. (Courtesy of Chicago Rawhide.)

Table 1.2 The Interaction of Seal and Application Variables

Seal variables	Effects causing static or dynamic leakage	Application variables
Type		
Material	Thermal expansion and contraction	Static or dynamic
Design	Seal material degradation	Sealed medium properties
	Corrosion	Type of shaft speed and motion
	Fatigue	Shaft eccentricity
	Vibration	Shaft diameter
	Wear	Shaft finish
	Shaft and bore defects	Shaft material
	Lubricant breakdown	Housing
	Case leakage	Finish
	Misalignment	Eccentricity
		Interference tolerance
		Operation
		Cycles (pressure, temperature)
		Run time
		Ambient
		Temperature and range
		Ozone
		Dust/mud
		Pressure and range

Source: Chicago Rawhide.

Some seals wear the shaft surface and/or the housing walls. These parts may require refinishing or replacement before a new seal is installed. Some seal systems may require buffer fluids, pumps, and filters. Operating costs should include the loss of buffer fluid, the power required to drive the pumps, and the cost to replace the filters.

Miscellaneous costs cover the loss of production when the machinery is idle while the seal is being replaced. Leaking seals can cause damage to other components if a fire results or if lubricant levels become low. Leakage of harmful fluids can create health hazards and/or damage the environment.

Dynamic shaft applications can be sealed with soft packings, elastomeric radial lip seals, and mechanical seals. For a 4 in. (101.6 mm) shaft, the initial costs of a heavy duty balanced mechanical seal is over 100 times that of packings or an elastomeric radial lip seal (Table 1.3). The mechanical seal may provide the longest life, but there may not be sufficient room in the application or enough money available. In some cases, the extra life is

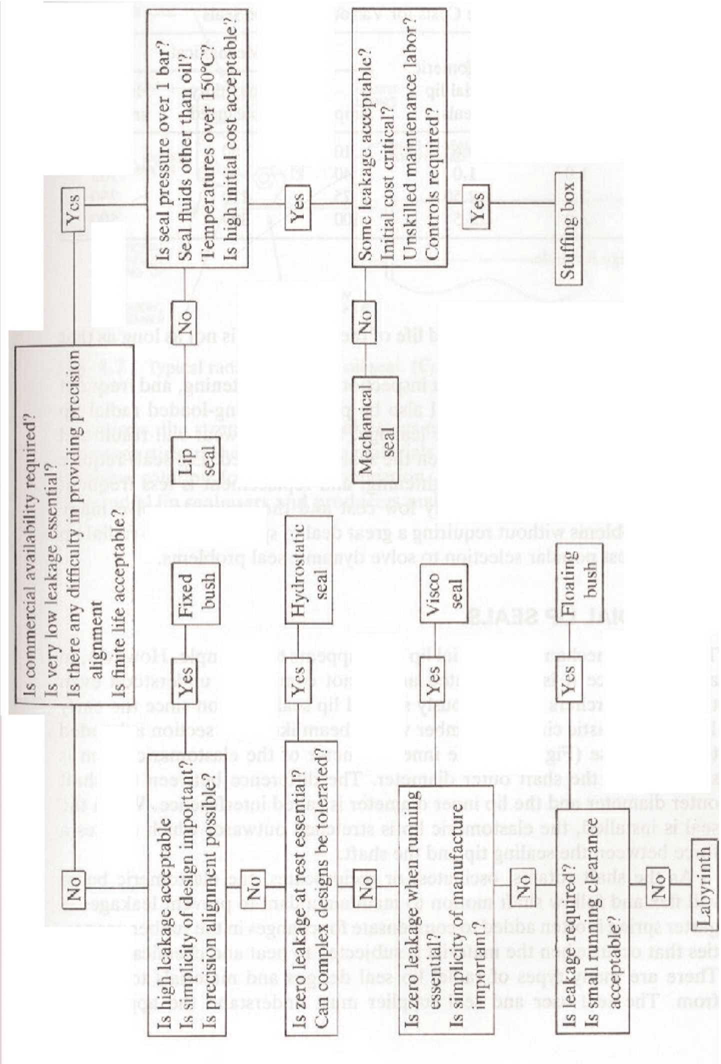
Is commercial availability required?
Is very low leakage essential?
Is there any difficulty in providing precision alignment
Is finite life acceptable?
Yes
No
Is seal pressure over 1 bar?
Seal fluids other than oil?
Temperatures over 150°C?
Is high initial cost acceptable?
No
Lip seal
Yes
Some leakage acceptable?
Initial cost critical?
Unskilled maintenance labor?
Controls required?
No
Mechanical seal
Yes
Stuffing box
Is high leakage acceptable
Is simplicity of design important?
Is precision alignment possible?
Yes
Fixed bush
No
Is zero leakage at rest essential?
Can complex designs be tolerated?
Yes
Hydrostatic seal
No
Is zero leakage when running essential?
Is simplicity of manufacture important?
Yes
Visco seal
No
Is leakage required?
Is small running clearance acceptable?
Yes
Floating bush
No
Labyrinth

Fig. 1.6 Seal Selection Chart for rotary application. (Courtesy of Chicago Rawhide.)

Table 1.3 Relative Purchase Costs for Various Types of Seals

Shaft diameter (in.)	Soft packing	Elastomeric radial lip seals	Mechanical		
			Simple	Heavy duty unbalanced	Heavy duty unbalanced
1	0.4	0.6	10	60	75
4	1.0	1.0	40	95	105
8	2.0	2.50	75	110	250
10	3.0	5.5	100	300	500

Source: Chicago Rawhide.

not needed because the expected life of the equipment is not as long as that of the seal.

Packings will require frequent inspection, gland tightening, and frequent replacement. Some leakage will also be present. Spring-loaded radial lip seals will minimize or eliminate leakage. Some shaft wear will result and the shaft may require repair when the seals are replaced. Lip seals require less inspection and no gland tightening, and replacement is less frequent than for packing. The relatively low cost and the ability to solve many sealing problems without requiring a great deal of space make the radial lip seal the most popular selection to solve dynamic seal problems.

1.3 RADIAL LIP SEALS

The sealing mechanism of radial lip seals appears to be simple. However, in actual practice it is complicated and is not completely understood even though researchers have seriously studied lip seal function since the early 1950s. An elastic circular member with a beamlike cross section is bonded to a rigid case (Fig. 1.7). The inner diameter of the elastomeric beam is smaller than the shaft outer diameter. The difference between the shaft outer diameter and the lip inner diameter is called interference. When the seal is installed, the elastomeric lip is stretched outward, which creates a force between the sealing tip and the shaft.

As the shaft rotates, oscillates, or reciprocates, the elastomeric beam will flex and follow shaft motion to maintain a dam to prevent leakage. A garter spring is often added to compensate for changes in the rubber properties that occur when the material is subjected to heat and chemical attack. There are many types of radial lip seal designs and materials to choose from. The seal user and seal supplier must understand the application

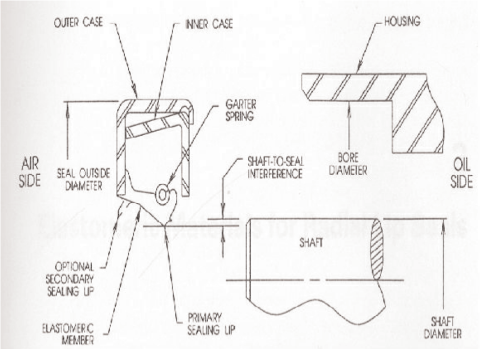
OUTER CASE
INNER CASE
HOUSING
GARTER SPRING
AIR SIDE
SEAL OUTSIDE DIAMETER
SHAFT-TO-SEAL INTERFERENCE
BORE DIAMETER
OIL SIDE
SHAFT
OPTIONAL SECONDARY SEALING LIP
ELASTOMERIC MEMBER
PRIMARY SEALING LIP
SHAFT DIAMETER

Fig. 1.7 Typical radial lip type of oil seal. (Courtesy of Chicago Rawhide.)

conditions, the strengths and weaknesses of the available seal designs, and the properties of the elastomeric materials in order to decide upon a cost-effective solution for a given application. This book provides a guide to help radial lip seal users and producers make the proper decisions.

2

Elastomeric Materials for Radial Lip Seals

2.1 POLYMER SELECTION

The heart of a radial lip seal is the elastomeric material or compound that is selected to form the sealing lip. This material consists of a base polymer that is mixed with fillers, antioxidants, curing agents, and other ingredients that enhance performance. The base polymer must be selected with care to ensure that the needs of the application are satisfied without sustaining excessive costs. The seal user and manufacturer must have an understanding of the base polymer properties, limitations, and strengths to ensure that the proper base polymer is selected for the application [77, 131, 132, 204, 206].

The base polymer will define the temperature range and fluid resistance of the compound. Proper compounding techniques will provide the flexibility the material requires to ensure that the seal lip will follow the dynamic movements and vibrations of the shaft [123]. After compounding, the material is placed in a mold and vulcanized or cured to the final shape of the seal. Vulcanization is an irreversible process and must be controlled carefully to achieve the correct physical properties. The material must be compatible with the sealed fluid, be abrasion- and tear-resistant, and maintain physical properties for an extended period of time.

2.1.1 Nitrile (NBR)

Nitrile (NBR) is widely used because it combines the lowest cost relative to other base polymers with good oil and abrasion resistance. The biggest drawback with NBR is a lack of heat resistance at elevated temperatures. The material will harden, crack, and lose interference with the shaft, which results in seal failure.

The practical temperature range for rotary shaft seals made with the NBR base polymer is -40 to +110°C (-40 to +230°F). At low temperatures, the material becomes stiff and may crack when the shaft is rotated. Nitrile is a copolymer of butadiene and acrylonitrile (ACN). The upper temperature limit can be extended by increasing the percentage of ACN, but the lower temperature limit is also increased. If the ACN percentage is reduced, the low-temperature limit is also reduced but the performance at elevated temperatures will suffer. Compounds made with an NBR base polymer are a good choice if the operating conditions are mild. Sulfur-cured NBR materials will harden quickly when exposed to gear oil EP additive packages at 110°C (230°F). Peroxide-cured NBR will give better results.

2.1.2 Saturated Nitriles (HNBR)

An advanced nitrile polymer (HNBR) has been developed that resists hardening and remains flexible at elevated temperatures up to 135°C (275°F). It provides abrasion resistance, oil and EP additive resistance, and good low-temperature performance. The use of this polymer in oil seals has been limited owing to manufacturing process difficulties and high cost. The base HNBR polymer is approximately 15 times as expensive as the conventional NBR material.

2.1.3 Polyacrylic Polymers (ACM)

Polyacrylic polymers cost approximately 2.5 times as much as NBR but offer better performance at elevated temperatures. Polyacrylics resist the EP additives in gear oils and are preferred over sulfur- and peroxide-cured NBR compounds. They also function well in engine and transmission fluids if the upper temperature is 135°C (275°F) or less. Polyacrylic is not recommended for seals exposed to water, acids, bases, and other polar solvents. Humidity can also affect the material properties [81]. Extreme care must be taken when selecting a polyacrylic for low-temperature applications. Some base polymers will become brittle and fracture at -15°C (5°F). Other polymers will function well at -40°C (-40°F) [22]. Polyacrylates also have poor dry running characteristics and will exhibit wide lip wear that will lead to early failures.

2.1.4 Ethylene Acrylic (AEM)

Ethylene acrylic (AEM) has properties very similar to those of polyacrylate materials [50]. It is a blend of ethylene and methyl acrylate and has better moisture resistance and lower-temperature resistance than polyacrylics. AEM materials rarely fracture at low temperatures. The temperature range is -40 to +150˚C (-40 to +302˚F).

2.1.5 Silicone (VMQ)

Silicone (VMQ) remains soft and flexible over a very broad temperature range (-70 to +150˚C; -94 to +302˚F). The polymer cost is about 6.25 times that of NBR. The primary use of silicone shaft seal materials used to be gasoline or diesel engine crankshaft seals and transmission seals. Modern oils with new additive packages can attack the silicone, causing the compound to decompose, which results in catastrophic failure. Silicone also has poor dry running characteristics that can result in large chunks tearing out of the lip during operation. Many times, silicone seals are soaked with oil before installation to prevent stick-slip in sparsely lubricated applications.

Silicone readily absorbs oil, and the seal lip can swell 30% or more. Seal designers often place heavy garter springs on silicone lips to limit the amount of interference loss when the material swells. Silicone seals cannot be used if they are exposed to gasoline, chlorinated oils, and aromatic solvents. The material tears easily and can often be damaged during handling. The problems and limitations of silicone materials has resulted in a decline in the use of this material for rotary shaft seal applications.

2.1.6 Fluorosilicone (FVMQ)

Fluorosilicone (FVMQ) is a fluorinated version of silicone and is very expensive (its cost is 22.5 times that of NBR). It has excellent oil and chemical resistance with lower swell than conventional silicone. It has excellent low-temperature properties. FVMQ is often used for static seals in fuel systems. The high cost and poor dry running properties result in limited use as a rotating shaft seal.

2.1.7 Fluoroelastomer (FKM)

Fluoroelastomer (FKM) is also an expensive material, costing about 22.5 as much as nitrile. It is being used in more and more rotating shaft seal applications because it has excellent oil and chemical resistance, which results in very low swell [234]. It has good high-temperature properties and excellent abrasion resistance. It will not fracture at low temperatures but will become very stiff and may not follow shaft wobble and vibrations until frictional heat warms the elastomer. This may cause sporadic leakage if synthetic lubricants that remain fluid at low temperatures are used. FKM compounds are recommended for high-speed, high-temperature operation with sparse lubrication.

2.1.8 Polytetrafluoroethylene (PTFE)

Polytetrafluoroethylene (PTFE) is a plastic material that is sintered into a billet at high temperatures (370°C; 700°F) and then machined into various shapes for seals. The polymer is approximately 25 times as expensive as NBR but is resistant to almost all fluids over a broad temperature range. It is easily damaged, and care must be exercised during handling and installation. Since the material is a plastic, it will not follow shaft wobble and vibrations very well. Special seal designs are required. PTFE seals are used primarily in high-speed, high-temperature applications with sparse lubrication.

2.1.9 Miscellaneous Polymers

Other polymers are used to make rotating shaft seals for specific applications [36, 77, 222]. The usage is minor and represents a small fraction of the oil seal population. Urethane polymers are often used to make seals in very dirty applications such as off-highway earthmoving equipment. These polymers are tough and wear-resistant but have poor high-temperature resistance. Butyl and ethylene propylene diene monomers (EPDM) are used to seal synthetic brake fluids and polar solvents. Epichlorohydrin is a polymer that has properties similar to those of polyacrylate and ethylene acrylics. It is not widely used in shaft seal applications because of manufacturing problems.

The characteristics of the common polymers used for rotary shaft seals are summarized in Table 2.1.

2.2 TESTING OF UNVULCANIZED ELASTOMERIC MATERIALS

The raw polymers are mixed with other ingredients and chemicals that will cause the polymer to cross-link when exposed to heat. This cross-linking process is known as vulcanization. The mixed material must be stored in a cool area to prevent premature vulcanization (scorch), which would make the material very difficult to process and mold. Unvulcanized material is very stiff at room temperature. Test methods to measure the viscosity, non-Newtonian flow behavior, and vulcanization characteristics of the material must use machines that are robust [71, 72].

2.2.1 Rotating Disk Viscometer (Mooney Machine)

The shearing disk viscometer was developed by Melvin Mooney in the 1930s and became the standard of the industry. It is still used today to measure material viscosity and scorch time. Scorch time is the amount of time the material can remain at a given temperature before it begins to vulcanize. As the test temperature is increased, the scorch time will drop. A low scorch time indicates that the material can be adversely affected by exposure to heat before processing.

The edges and surfaces of a cylindrical metal disk are grooved to grip the rubber sample and prevent slippage when the disk is rotated. The disk sits in a chamber that is filled with rubber (Fig. 2.1). One piece of the rubber sample is placed under the rotor, and the other is placed on top (Fig. 2.2). The cavity is then closed, and the rubber is squeezed with a force of 2500 lb. The sample is warmed for 1 min, and then the disk is rotated at 2 rpm in one direction only. The torque required to turn the rotor is measured relative to time and can be related directly to the material viscosity. The shear rate is quite low (approximately 2 sec^{-1}) and is applicable for those processes (i.e., compression or some transfer molding operations) that have low shear rates. Other methods of measuring viscosity must be used for higher shear rate processes such as extrusion and injection molding.

The viscosity (recorded in Mooney units) drops as the material warms up, reaches a plateau, and then increases steadily as the material begins to vulcanize. A shear pin on the rotor drive mechanism is used to protect the instrument from becoming overloaded as the material becomes stiffer. An analysis of a typical curve (Fig. 2.3) reveals some valuable information. The minimum viscosity (MV) is the baseline and is approximately 20 Mooney units for the rubber sample used to generate the curve. T_5 is the time required for the Mooney reading to increase 5 units over MV (27 min in this case). T35 is the time required for the Mooney reading to increase 35 units over MV (34 min in Fig. 2.3), and TΔ30 is defined by Equation (2.1) and is known as the cure index (7 min). The Mooney rise readings can be recorded at various other points as required.

$$T_{\Delta 30} = T_{35} - T_5 \qquad \text{(minutes)} \qquad (2.1)$$

TABLE 2.1 Polymer Selection Chart for Rotary Shaft Seals

Material	Approx. relative cost (polymer)	Approx relative cost (seals)	Sump oil temperature ˚C	Sump oil temperature ˚F	Advantages	Disadvantages
Nitrile (NBR)	1	1	-40 to 110 (depends upon ACN content)	-40 to 230 (depends upon ACN content)	Low cost, low swell. Good wear and oil resistance at moderate temp.	Poor resistance to EP additives. Poor high-temp resistance.
Saturated nitrile (HNBR)	15	1.75	-40 to 135	-40 to 275	Good wear and oil resistance at elevated temperature. Good low temperature properties.	High cost. Manufacturing process problems.
Polyacrylic (ACM)	2.5	1.15	-40 to 135	-40 to 275	Good oil resistance. Low swell. Generally resistant to EP additives.	Fair wear properties. Poor dry running. Poor water resistance. May crack at low temp.
Ethylene acrylic (AEM)	2.5	1.15	-40 to 150	-40 to 302	Good temperature range. Good moisture resistance. Fair abrasion resistance.	Poor dry run characteristics. May have high swell in some fluids.

Silicone (VMQ)	6.25	1.3	-70 to 150	-94 to 302	Very broad temperature range.	Poor dry running properties. Poor resistance to oxidized oil and some EP additives.
Fluorosilicone (FVMQ)	22.5	2	-70 to 150	-94 to 302	Excellent oil and chemical resistance. Good low-temperature properties.	Poor tear strength. Poor dry running. Poor wear resistance.
Fluoroelastomer (FKM)	22.5	2	-40 to 150	-40 to 302	Excellent oil and chemical resistance. . Good wear properties. Low swell	Becomes stiff at low temperatures. Poor followability at low temperatures.
Polytetrafluoroethylene (PTFE)	25	2.5	-80 to 200	-112 to 392	Excellent oil and chemical resistance. Excellent temperature range.	Easily damaged. Becomes stiff at low temperature. High cost. Poor followability at low temperature.

Source: Chicago Rawhide.

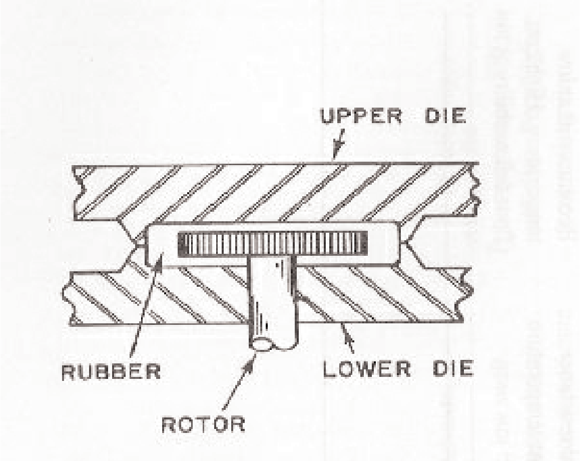
UPPER DIE
RUBBER
LOWER DIE
ROTOR

Fig. 2.1 Mooney rotor and chamber.

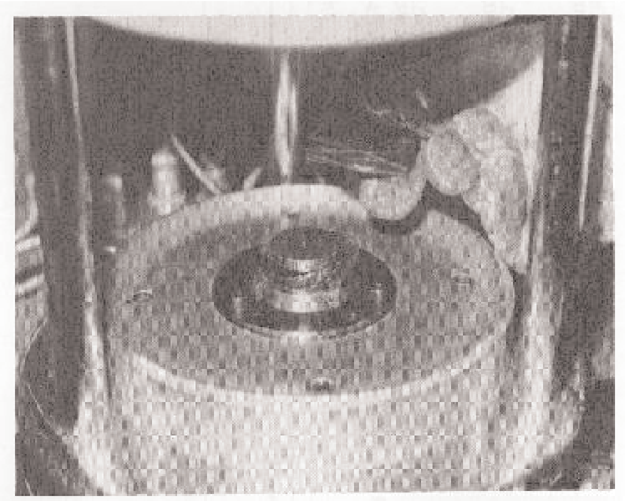

Fig. 2.2 Loading rubber samples into the Mooney chamber.

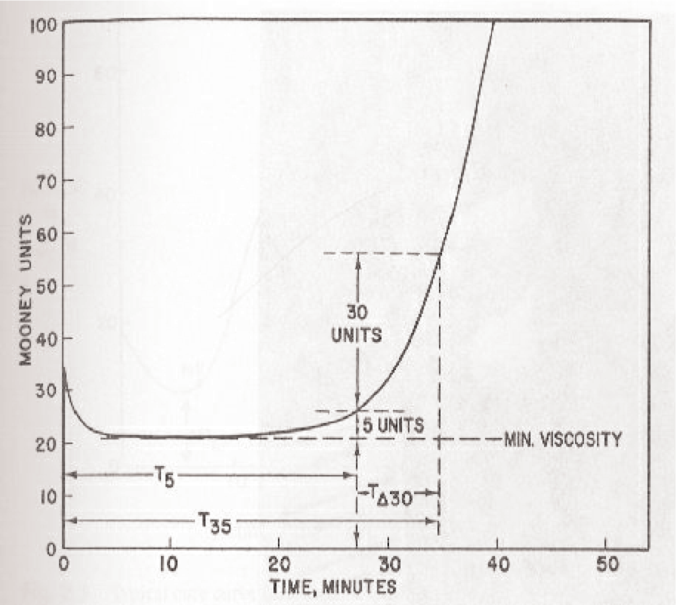

Fig. 2.3 Typical Mooney scorch curve. (Courtesy of Chicago Rawhide.)

The test is described in ASTM D1646 and is used as a quality check to ensure material consistency. The test is also used by researchers developing new materials. A typical machine is shown in Fig. 2.4.

2.2.2 Oscillating Disk Rheometer

The Mooney machine is limited because either the sample will tear and slip or the shear pin will break as the material cures and becomes stiff. The machine cannot provide data at a point in time well after the scorch point has been reached. This problem is directly related to the unidirectional rotation of the rotor. Machines known as curemeters use a disk that oscillates in the chamber and measures the torque during vulcanization and after vulcanization is complete. The torque will drop initially as the sample warms up, reach a plateau, and then increase as vulcanization begins. Since the rotor oscillates, a maximum torque value will be reached when the rubber is fully cured. Some rubber materials may become overcured as heat continues to be applied during the test, and the torque may decline as the material deteriorates.

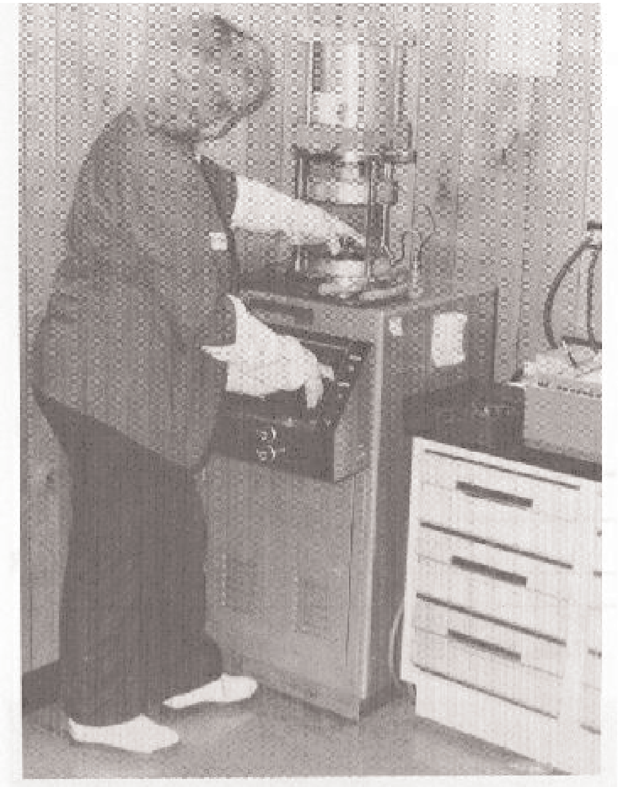

Fig. 2.4 A rotating disk viscometer (Mooney machine).

A typical rheometer curve appears in Fig. 2.5. The minimum torque is M^L, the maximum torque is M_{HF}, the scorch time is t_{s2}, and the time to reach a 90% cure level is $t_c(90)$. The test procedure is described by ASTM D2705 and is used for material development and quality control. A typical rheometer is shown in Fig. 2.6.

2.3 TESTING VULCANIZED ELASTOMERIC MATERIALS

The sealing member of a radial lip seal is molded from an oil-resistant elastomeric or rubber compound. Rubber is very different from other materials. It can be stretched to several times its original length without breaking and will return to its original configuration when the stretch forces are released. Steel and other metallic substances require a great deal of force (stress) to stretch (strain) the material. The relationship between stress and strain is linear for metallic materials until the yield point is reached. The

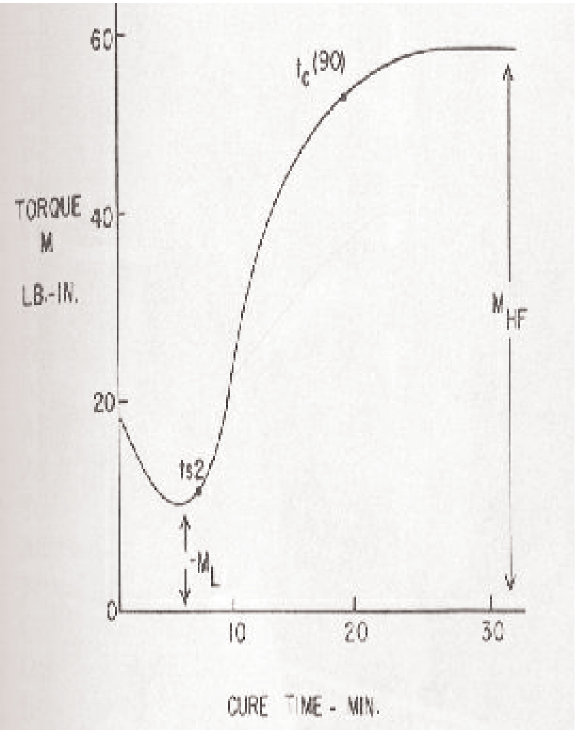

Fig. 2.5 Typical cure curve measured by oscillating disk rheometer. (Courtesy of Chicago Rawhide.)

slope of the stress-strain curve for a metal is constant and is known as Young's modulus. The relationship between stress and strain for elastomeric materials is nonlinear; the slope is not constant, and a modulus value is meaningless without specifying the amount of strain.

A rubber sample may not fully recover its original height after being compressed. The loss is known as compression set, and this property is unique to elastomeric materials. There are many other unique physical properties that are used to characterize rubber compounds. The original physical properties of rubber will change when the rubber is exposed to heat, oil, chemicals, sunlight, ozone, weather, and other outside influences. The original physical properties and the resistance to change of these properties can be altered dramatically by compounding the basic polymer with various fillers, antioxidants, plasticizers, curative agents, etc.

Unique methods have been developed for measuring the original physical properties and aging characteristics of elastomeric compounds. Seal designers, manufacturers, and users should all have an understanding of the tests used to characterize an elastomer. The tests have limitations because most are not directly related to radial lip seal performance. The tests can only be used as a screen to eliminate those compounds that are grossly

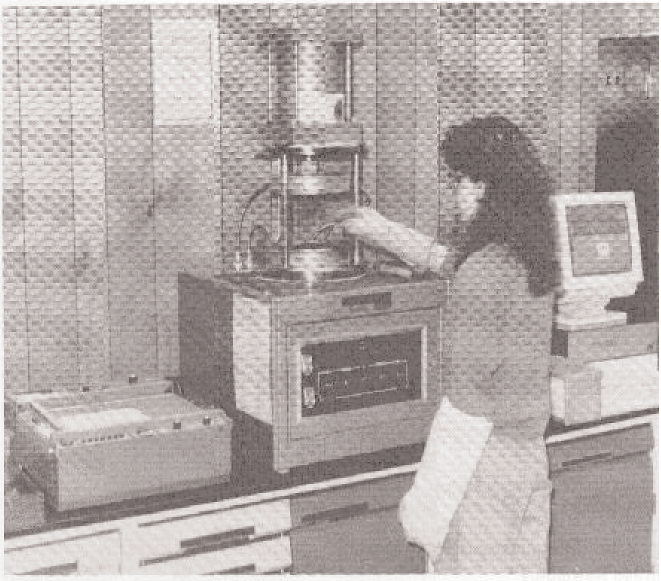

Fig. 2.6 Oscillating disk rheometer.

inadequate for the application. Physical property tests can also be used for quality control to measure the consistency of rubber materials. A list of the typical ASTM (American Society for Testing and Materials) tests for vulcanized elastomers appears in Table 2.2.

2.3.1 Hardness (ASTM D2240 and D1415)

There are many devices used to measure the hardness of materials. For elastomeric materials, hardness is defined as the relative resistance of the sample surface to indentation by a ball of specific dimensions subjected to a specified load. Elastomeric materials require special instruments and scales because they are soft relative to most other materials. The instrument scale represents the depth of penetration of the indenter into the sample. Arbitrary units range from 0 to 100, with 100 being the hardest. Relatively thick sample slabs are often molded by compounders and are used to prepare samples for various tests. A hand-held instrument known as a Shore A durometer (Fig. 2.7a) is often used to measure the hardness of the thick test slabs or thick samples obtained from the finished product. The in-

Table 2.2 ASTM Test Methods for Vulcanized Elastomers

Test	ASTM number
Hardness--;Shore durometer A	D2240
Hardness--Wallace microhardness	D1415
Tensile strength, tensile stress (modulus), and elongation	D412
Compression set at elevated temperature	D395
Compression set at low temperature	D1229
Flex fatigue	D430
Flex fatigue, crack growth	D813
Abrasion resistance (NBS abrader)	D1630
Abrasion resistance (Pico abrader)	D2228
Abrasion resistance (Taber abrader)	D3389
Tear strength	D624
Adhesion	D413, D429
Permeability	D814, D815
Cold test procedures	D1053, D2137, D1329
Ozone resistance	D1149, D470, D1171
Heat aging	D573, D865
Oil aging	D471
Compression deflection resistance	D575
Resilience	D945

denter point projects from the flat bottom of the case and is held at the zero point by a spring. The indenter point is pushed back into the case against the spring when the instrument is placed against the rubber sample. This action creates a rotation of the dial. Hard elastomers will give more rotation and higher readings than soft elastomers.

The Shore instrument cannot be used to measure the hardness of finished seal products because typical seal sample cross sections and thicknesses are too small. A Wallace microhardness tester with a very small indenter is used to measure these thin (1.0 mm) rubber samples (Fig. 2.7b). The scale ranges from 0 to 100, and the units are referred to as international hardness degrees. Zero corresponds to a material with an elastic modulus of zero, and 100 would be a material with an infinite elastic modulus (ASTM D1415). Shore A values of rubber slab samples will typically be five to ten points higher than a Wallace microhardness measurement on the same slab. Microhardness values for typical shaft seal materials can range from a low of 65 to as high as 90. The units typically have a tolerance of ±5 points. A typical specification for a shaft seal material might be 75 ± 5.

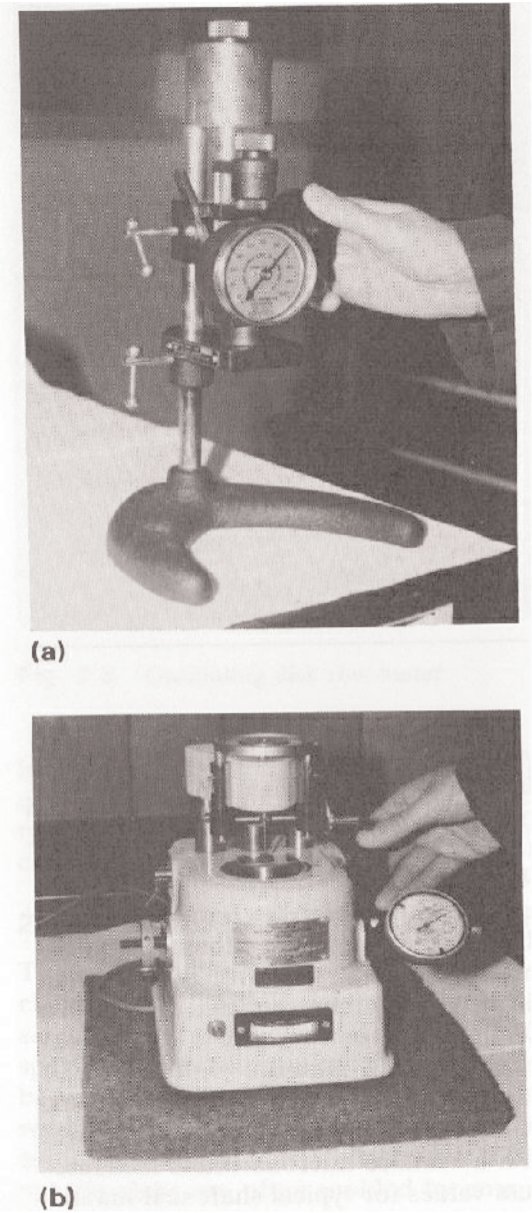

Fig. 2.7 Measuring hardness of vulcanized elastomers.

Wallace Microhardness Units

The Wallace instrument (Fig. 2.7b) can be used to determine the hardness of samples obtained from the lip of an oil seal. The sample must be prepared carefully, and the procedure is described in detail in Ref. 152.

Samples about 6 mm (0.25 in.) long are cut from the seal lip and are glued to a round steel or glass mounting disk with a thin layer (0.05 mm; 0.002 in.) of liquid adhesive (Fig. 2.8). The mounting disk should be 25-38 mm (1-1.5 in.) in diameter and about 6.3 mm (0.25 in.) thick. Samples from slabs can also be glued to the disk (Fig. 2.9). These samples should be approximately 6 mm (1/4 in.) square. The microhardness indenter can be applied directly to the surface of the slab samples or in the spring groove of the lip seal sample (Fig. 2.9).

The hardness specification for shaft seal materials has no direct relationship to seal performance because seal function cannot be related to the depth of penetration of a probe on the seal surface. In addition, the readings themselves are often suspect, as they are highly dependent upon operator technique and interpretation. Machine-to-machine variations can also be extreme even though standards for calibrating and using the instruments have been established. Both seal users and manufacturers continue to specify, measure, and reject seal materials and finished products even though

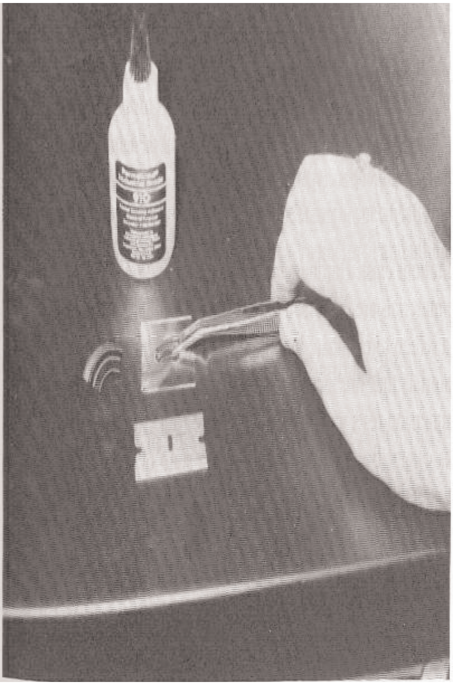

Fig. 2.8 Mounting seal samples for microhardness measurements.

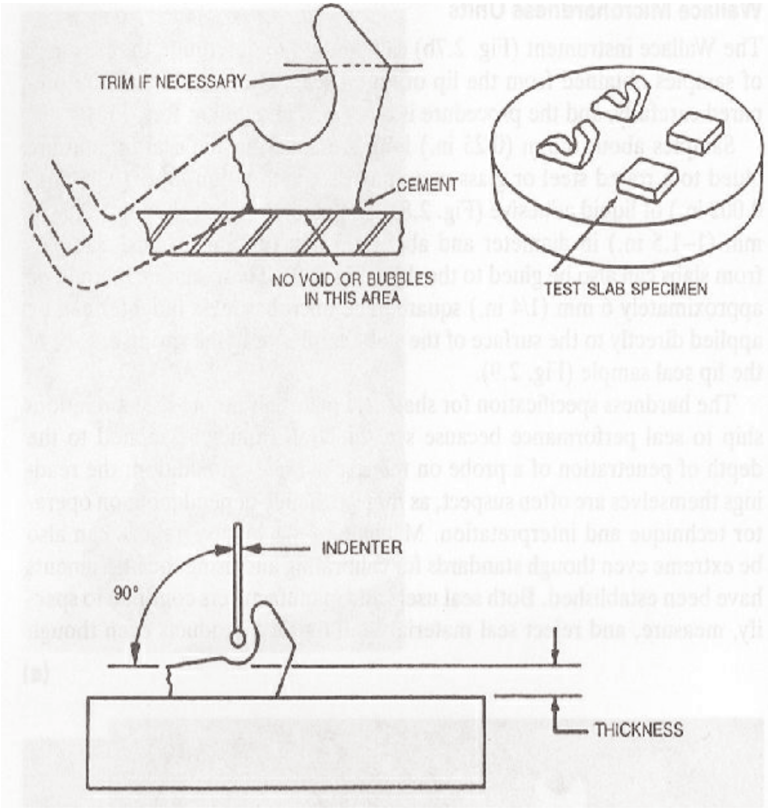

Fig. 2.9 Preparation of samples for microhardness measurements. (Courtesy of Rubber Manufacturers Association.)

these limitations exist. Hardness measurements and their change with time when the materials are immersed in test fluids at elevated temperatures can serve as a screen to eliminate materials that are incompatible with the test fluid. Materials that are incompatible will become too hard (or too soft) during immersion tests within a short period of time. Microhardness measurements cannot be relied upon to give the state of cure of finished oil seals. The differential scanning calorimeter (DSC) has been used to determine how well elastomers have been vulcanized (122).

2.3.2 Tensile Strength, Tensile Stress (Modulus), and Elongation (ASTM D412)

Most rubber compounds have specifications that include the required tensile strength and elongation at break as well as the tensile stress or modulus at specified levels of stress or elongation. Tensile strength is defined as the force per unit area of the original cross section that is being applied to stretch a dumbbell-shaped sample at the time it ruptures. The elongation or strain is the amount of stretch or extension expressed as a percentage of the original length when a given force is applied to the ends of the dumbbell-shaped sample. Ultimate elongation is the amount of stretch that exists when the sample ruptures. The tensile stress or modulus is the force per unit area of original cross section required to stretch the rubber sample a given amount. Since the stress-strain curve for rubber materials is nonlinear, Young's modulus is meaningless, and any modulus number quoted for rubber must include the percent elongation. The numbers are typically reported as 100% modulus, 200% modulus, etc., which simply means the modulus at 100% elongation, 200% elongation, etc. The stress-strain curve is developed by mounting a dumbbell-shaped sample that is die cut from slabs about 1-2 mm (0.04-0.08 in.) thick between the jaws of a test machine (Fig. 2.10) that will elongate the sample at the rate of 50 cm/min (20 in./min) until breakage occurs. The dumbbell specimen is about 10-13 cm (4-5 in.) long. Typical values for an elastomeric stress-strain curve appear in Table 2.3.

Unfortunately, the stress-strain curve and the modulus specifications have little to do with oil seal function. Most seal lip designs are subjected to small amounts of elongation (3-5%) when the seal member is placed over the shaft. In service, the seal is never elongated 100, 200, 300%, or until breakage occurs. Many seals users insist on specifying high values for tensile strength and elongation that may hinder the development of a compound that has more useful sealing properties at a lower cost. It is possible for a compounder to develop many different materials that will meet the same tensile strength/elongation specifications. Some of these compounds will not function as an oil seal. A more meaningful specification would be a 5-10% modulus that can be related to seal radial force and ultimate performance. These specifications do not currently exist.

2.3.3 Compression Set (ASTM D395 and D1229)

Rubber materials may not recover their original dimensions after they have been stretched, compressed, or sheared for a period of time. This loss is known as permanent set, and test procedures have been defined to measure this property under each of the three conditions of load: tension, compression, and shear. The test most commonly used for seal materials is to determine compression set, which is described by ASTM D395 for elevated temperatures and by ASTM D1229 for low temperatures. A pill of material with standard dimensions is molded or built up from thin sections that are die cut from a sample slab. The pill is placed between two steel plates and

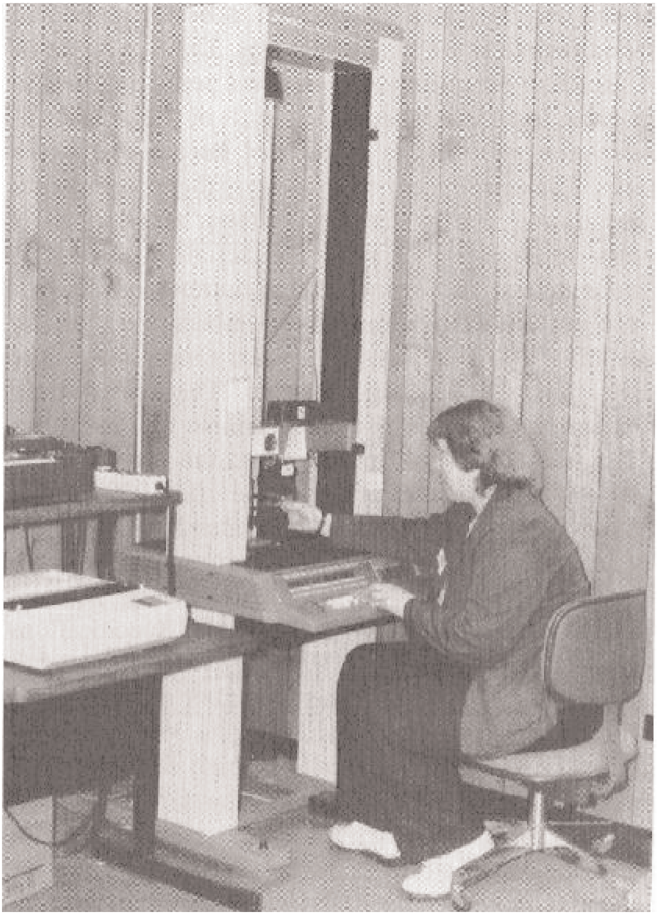

Fig. 2.10 Test machine for determining elongation at break.

compressed to 75% of its original thickness (Fig. 2.11). The test assembly is placed in an oven at a specified elevated temperature for a specified time period. The pill is removed from the test assembly, and the thickness is measured at room temperature. Compression set is the difference between the original thickness of the pill and the thickness of the pill after the test. It is defined as a percentage of the original deflection:

$$\text{Compression set}\ (\%) = 100\left(\frac{t_b - t_a}{0.25}\right) = 400\left(1 - \frac{t_a}{t_b}\right) \qquad (2.2)$$

TABLE 2.3 Typical Stress-Strain Curve for an Elastomer

Strain (% elongation)	Stress at given elongation (MPa)
100	0.4
200	1.0
300	1.75
400	2.75
500	4.25
550[a]	8[b]

[a] Sample broke at 550% elongation.

[b] Tensile strength is 8 MPa.

Source: Chicago Rawhide.

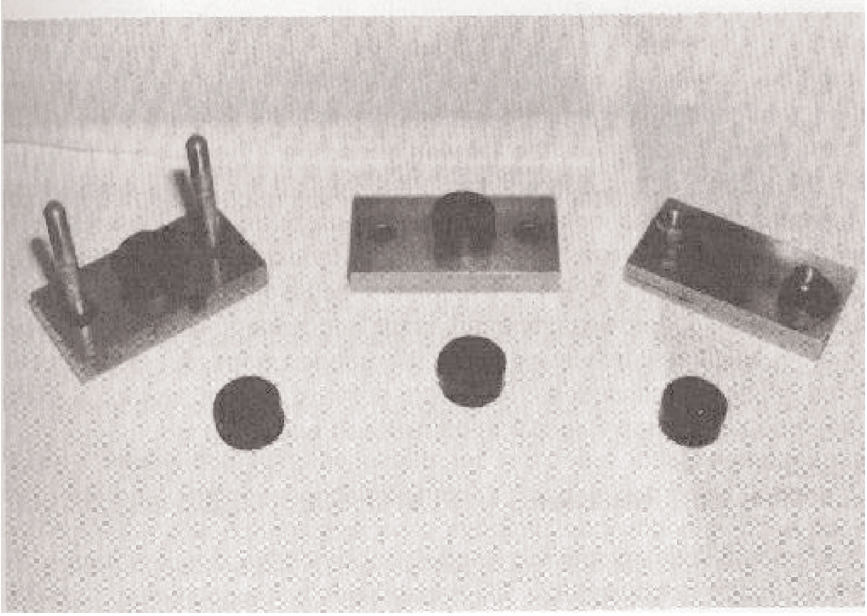

Fig. 2.11 Measuring compression set at constant deflection.

The low-temperature test is identical except that a freezer is employed and the pill thickness after test is measured at the low temperature. This reading is actually not a permanent set; some recovery may result as the sample warms up to room temperature.

The compression set test is an excellent test procedure for materials used in O-rings, gaskets, and other static seals that must function under compression in actual use. It does not have much value for materials used in lip seals for dynamic applications; lip seal material is never under compression. The procedure is of value for those seals that have rubber gaskets molded at the seal outside diameter that form a static seal with the housing bore after installation. Compression set is almost universally specified for dynamic seal compounds even though it has little functional value.

2.3.4 Flex Fatigue (ASTM D430 and D813)

Materials that are subjected to repeated distortion from bending, extrusion, or compression often develop surface cracks that propagate and eventually result in rupture. Several test machines exist that subject samples to repeated motion for evaluating resistance to flex fatigue (ASTM D430). The DuPont and DeMattia machines are used primarily to study the growth of surface cracks. In addition, the DeMattia machine (Fig. 2.12) is used to study the growth of an induced crack (ASTM D813). These methods of determining the resistance to flex fatigue are useful for seal materials because the lip member is constantly being flexed, stretched, and relaxed as the shaft rotates. Care must be taken to compare the results of an experimental material to that of a known compound, as the interpretation of results can be subjective. Many researchers will test until rupture occurs and compare the number of cycles achieved to the failure point of known materials. Flex resistance is not a normal specification for shaft seal materials.

2.3.5 Abrasion Resistance (ASTM D1630, D2228, and D3389)

Many dynamic seals operate in a sparsely lubricated or dry condition with external contaminants present. The rubber must be tough enough to resist abrading under these conditions. Three test methods exist to evaluate the abrasion resistance of rubber compounds.

1. The NBS abrader (ASTM D1630) consists of a rotating drum with a standard abrasive paper attached to it. The sample is pressed against the paper for a specified time under a standardized load.
2. The Pico abrader (ASTM D2218) has a pair of tungsten knives that rotate and scrape the surface of the test material in the presence of an abrasive dust.

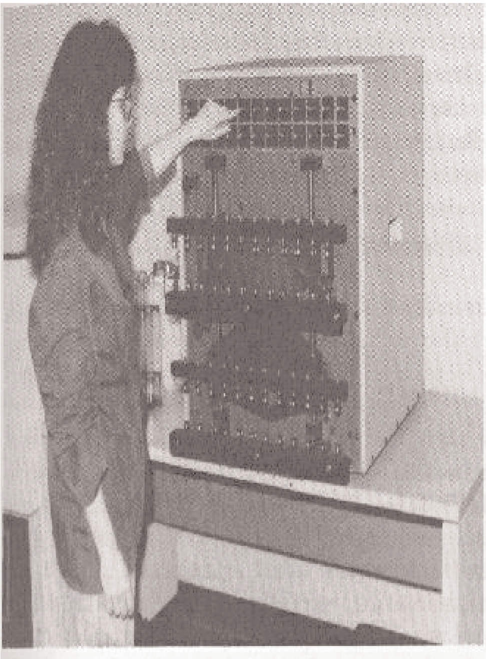

Fig 2.12 DeMattia test machine for studying flex fatigue.

3. The Taber abrader (ASTM D3389) has two rotating abrasive wheels that grind against the surface of the specimen. The weight or volume of the rubber sample after test is compared to the weight or volume before test to determine how much material was lost.

It is recommended that the results be compared to a known standard material to evaluate how well the sample compound performs. It should be noted that rubber compounds with excellent abrasion resistance do not always make good seals. Abrasion resistance tests are only one method of evaluating compounds and are used as a supplement to other material tests. Abrasion resistance does not usually form part of a shaft seal material specification.

2.3.6 Tear Strength (ASTM D624)

The tear strength of a compound will not form part of the material specification, but it is useful to know the resistance to damage during molding, handling, and use. The tear strength procedure described in ASTM D624 will give a method of determining what materials may have poor resis--

tance to handling. The test samples can be prepared with a standard nick or without nicks. A force is then applied, and the tear resistance is defined as that force per unit of sample thickness that will propagate the nick or cause tearing in a direction perpendicular to the direction of the applied force. Readings should be compared to results obtained from a standard compound.

2.3.7 Adhesion (ASTM D413 and D429)

Seal lips are often bonded directly to a metal or plastic case. The strength of the bond between the rubber and the metal or plastic is a critical parameter for the proper functioning of the oil seal. Laboratory tests are often conducted to measure the bond strength by stripping or peeling a rubber sample from the substrate. Other methods include a force applied normal to the plane of the bond or by shearing the rubber with forces parallel to the plane of the bond. The peel test is most commonly used to evaluate bonding systems for shaft seals. The metal coupon is prepared in a manner that simulates production procedures, and the adhesive is applied. The rubber is then bonded and vulcanized to the sample in a mold. The molding procedure must also simulate production procedures, as bond strength is highly dependent upon manufacturing techniques. A tensile strength testing machine is used to elongate the rubber at a constant speed, and the loads applied to the metal and rubber ends of the sample are measured. The force that causes the material to peel away from the metal can then be determined. In some cases, the rubber will rupture before the bond fails. This procedure cannot be used to evaluate the bond strength of finished seal products.

A subjective test has been used by seal manufacturers. The user grips the seal lip with a pair of pliers and pulls axially until the rubber ruptures or bond separation occurs. If a layer of rubber remains on the metal interface, then the bond is considered adequate. If the metal face shows areas that are clean of rubber, then the bond is poor. This method is highly dependent upon operator technique because the method of pull is uncontrolled. Some operators pull steadily, others will jerk at the material, and still others will pull and twist at the same time.

The Rubber Manufacturers Association has developed and published a procedure that allows quantitative analysis of the bond strength of finished oil seal products (162). This procedure describes a fixture that allows the outer shell of the seal to be clamped in the lower portion of a tensile tester. A set of jaws is attached to the upper portion of the tensile tester and clamped onto the rubber lip. The jaws are then moved upward at a rate of 500 mm/min (20 in. min) until the rubber ruptures or the bond has failed.

The load is recorded on a strip chart, and the peak load represents the point of failure. If the rubber fails before the bond, then the bond is acceptable. If the bond fails before the rubber ruptures, then a minimum acceptable load must be determined empirically. This minimum load will depend upon application factors, the rubber tensile strength, and experience. This method may be used for quality control, and the peak value at break can be charted. The manufacturing processes must be reviewed if the peak value at break falls below the minimum acceptable level (Fig. 2.13).

2.3.8 Permeability

Permeability is defined as the amount of fluid or gas that passes through a membrane. The units are typically reported as weight per unit time per area of exposed sample [g/sec-m^2)]. The fluid or gas typically diffuses through the

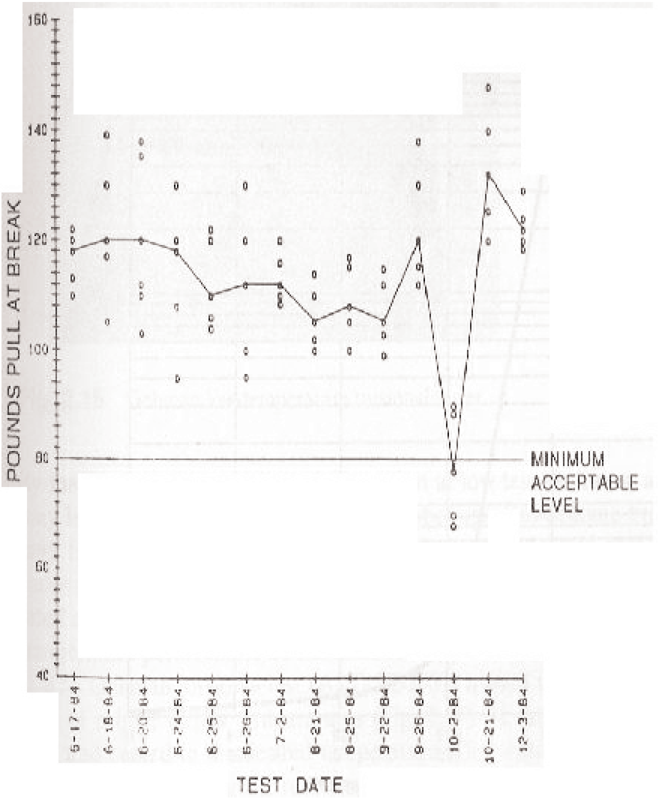

Fig. 2.13 Typical quality control chart for adhesion. (Courtesy of Rubber Manufacturers Association.)

film of rubber and is evaporated or degassed when it reaches the air surface on the outside of the rubber sample. The permeability of volatile fluids is determined by filling a jar with the test fluid, capping the jar with a thin membrane of the test elastomer of known area and thickness, inverting the jar, and measuring the fluid weight loss over a period of time (ASTM D814). The permeability of gases through the elastomer is an important measure for gaskets, oil rings, and other static seals that are required to retain gas. This parameter is not important for shaft seal materials designed for oil and grease applications. The method of determining permeability of gases through elastomers is described in ASTM D815.

2.3.9 Cold Test Procedures (ASTM D1053 and D2137)

When some elastomers are exposed to cold temperatures, they become stiff and lose their elasticity (Fig. 2.14). Seal materials that lose their elastic-

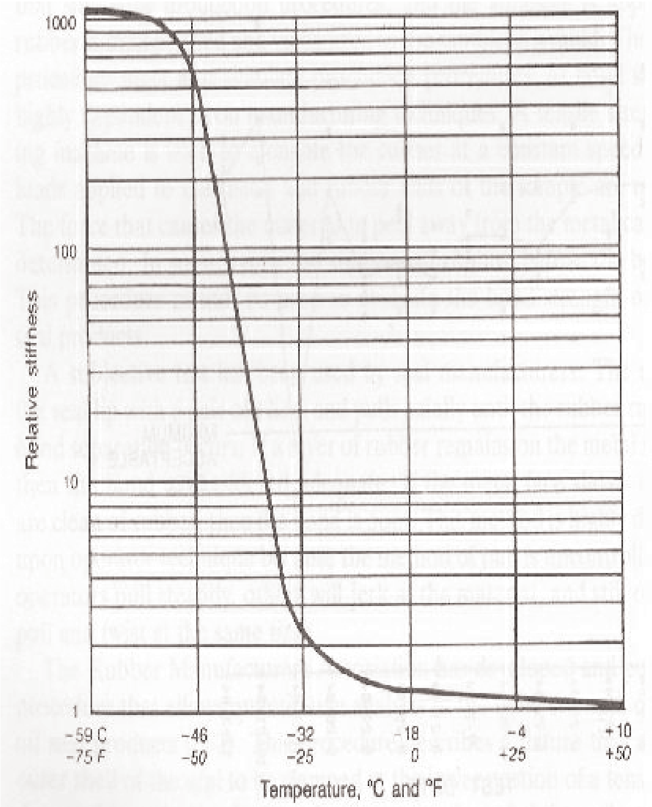

Fig. 2.14 Elastomers may become stiff at low temperature. (Courtesy of Rubber Manufacturers Association.)

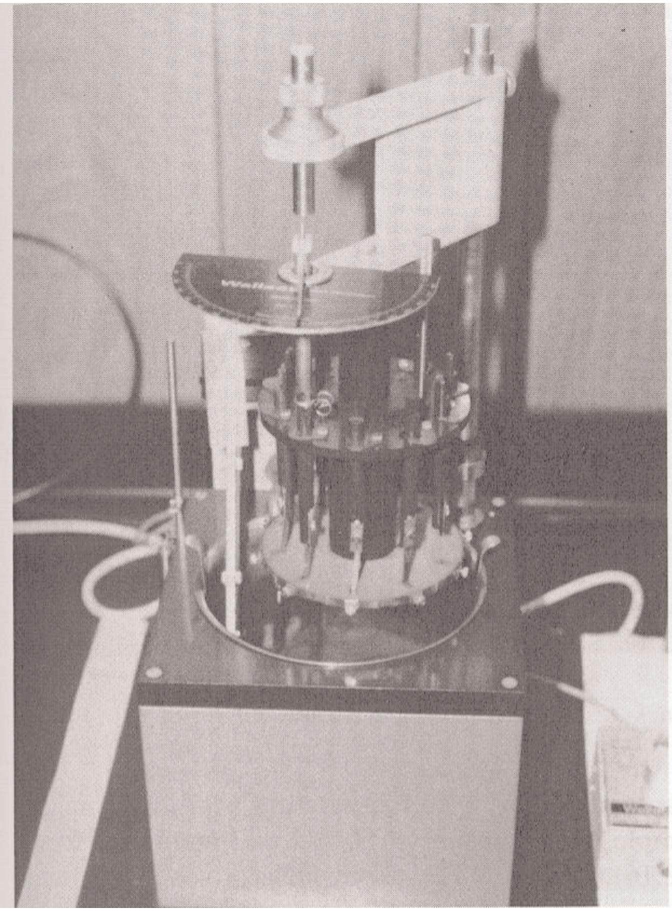

Fig. 2.15 Gehman low-temperature torsional tester.

ity may not be able to follow shaft motion at low temperatures, and fluid may leak until the rubber warms up. Rubber can also become brittle and may fracture when subjected to sudden impact. Fractured seal lips will cause catastrophic failure. Rubber stiffness and brittleness cannot be correlated at low temperatures. Two separate tests are used to measure the cold-temperature performance of elastomeric compounds.

The Gehman torsional test (ASTM D1053) is used to measure the stiffness of rubber at low temperatures (Fig. 2.15). A thin sample is twisted 180° and cooled to the desired temperature. The sample is then released, and the amount of angular recovery is measured after a 10 sec interval. A material with a larger amount of recovery is not as stiff as a material with a small amount of recovery and will follow shaft dynamics better than the stiffer material.

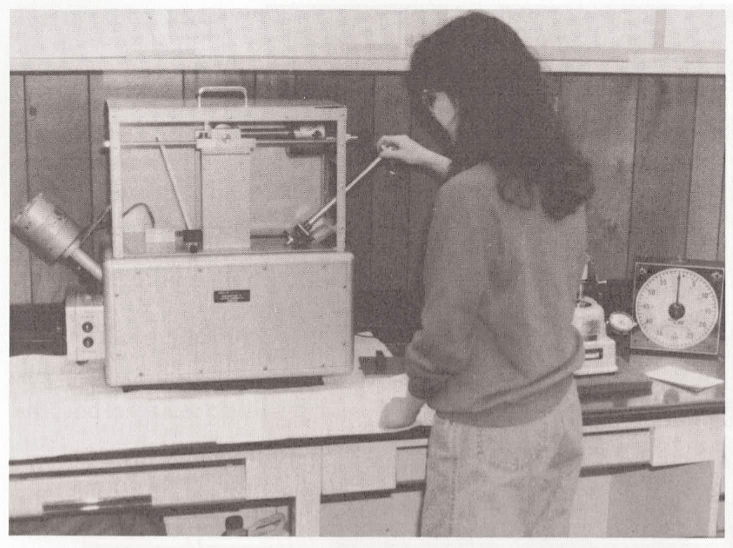

Fig. 2.16 Impact brittleness tester for cold temperatures.

The impact brittleness test measures the temperature at which a rigidly mounted and cooled sample breaks when struck by a known impact force (ASTM D2137) (Fig. 2.16). This test correlates well to seal functional breakage tests [159].

It is essential that the cold-temperature properties of oil seal elastomerics be considered when selecting a seal design and material for an application in which the seal is exposed to low temperatures.

2.3.10 Ozone Resistance (ASTM D470 and D1149)

Many rubber materials under stress will crack when exposed to atmospheric ozone over an extended period of time. Nitrile and some polyacrylate oil seal polymers are susceptible to attack from ozone. Silicones and fluoroelastomers are not affected. This attack is dramatically reduced when the seal is immersed in oil and installed in equipment that shields the seal from sunlight and atmosphere. Ozone damage can still result if the seal is exposed to excessive levels of ozone before installation or during use. Welding operations generate excessive quantities of ozone, and seals in the immediate area can be permanently damaged.

The relative ozone resistance of rubber samples can be determined by placing the samples in an ozone chamber for a specified length of time at a temperature of 104˚F (40˚C). One method (ASTM D1149) exposes a clamped, elongated (20%) sample to an ozone level of 50 ± 5 ppm in air. The samples are examined periodically for signs of cracking. A more severe test (ASTM D470) exposes a sample that has been wrapped around a mandrel to ozone levels of 100-150 ppm. The time for the first cracks to appear is recorded as the failure point.

2.3.11 Heat Aging (ASTM D573)

Oil seals are often exposed to elevated temperatures due to frictional heat generated by the operating equipment or by the power plant itself. Excessive heat may cause the material to become brittle, which results in a shortened life. Brittle and hard seal materials cannot respond to shaft dynamics, gaps will appear between the lip and the shaft, and leakage will result. Procedures for heat-aging rubber samples in an air oven (Fig. 2.17)

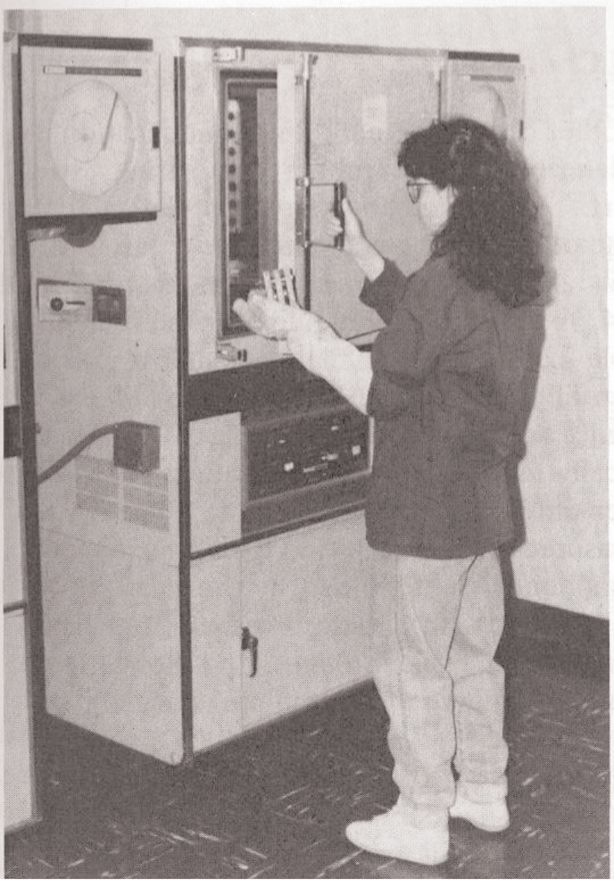

Fig. 2.17 Air-aging ovens.

appear in ASTM D573. The material hardness, modulus values at various elongations, tensile strength, and elongation at break are measured at various time intervals at the specified temperature and compared to the original properties.

The compression set is often measured also. The heat resistance of an elastomer is defined as its ability to resist permanent changes in its original properties. The air-aging test temperature should not be elevated beyond the service temperature in an effort to reduce test time. Excessive temperatures may cause a mode of deterioration that does not exist at the service temperature. Erroneous conclusions may cause the researcher to specify an expensive material that is not needed.

The rate at which elastomers deteriorate increases logarithmically with temperature. Small temperature changes can cause large changes in the rate of rubber deterioration. The oven controls must be adjusted to provide a steady uniform temperature with no hot spots or broad swings during the test. It is recommended that heat-aging tests of new compounds include samples of a standard with a known heat resistance.

2.3.12 Aging in Oils and Chemicals

Exposure to oils, water, and other chemicals found in the application can cause rubber materials to deteriorate. The rubber can shrink, swell, soften, harden, crack, or revert. The method and amount of deterioration depends upon the material, the oil, and test temperature and duration. A general guide to the chemical resistance of oil seal polymers to various chemicals appears in Table 2.4. A more complete guide for the four main polymer types is given in Table 2.5. Samples of the elastomer are placed into a test tube, which is then filled with oil and sealed (ASTM D471). The tubes are then placed into an aluminum block (Fig. 2.18) that is heated and controlled electrically to the desired temperature. The tubes are removed periodically (after times ranging from 24 hr to 30 days), and the physical properties are measured and compared to the original properties. Deterioration of the compound is determined by the changes that have occurred in its physical properties. In addition to the normal hardness, modulus at various elongations, tensile strength, and elongation at break, the change in weight and/or volume of the sample is usually measured. Elastomers often absorb the test fluid and swell, which also softens the material and reduces hardness during the early stages of the immersion test. The material will become harder again as the soak time is extended. In some cases, the test fluid will extract plasticizers and other ingredients contained within the elastomer, which causes the material to shrink and become harder.

(Text resumes on page 62)

Table 2.4 Elastomer Compatibility in Various Fluids

Medium	Nitrile	HNBR	Poly-acrylate	Ethylene acrylic	Silicone	Fluoro-silicone	Fluoro-elastomer	PTFE
Engine oil	Good	Good	Good	Good	Good	Good	Good	Good
ATF-A	Fair	Good	Good	Good	Good	Good	Good	Good
Grease	Good	Good	Fair	Fair	Fair	Fair	Good	Good
EP Lube	Fair-Poor	Good	Good	Gair	Poor	Fair	Good	Good
SAE 90	Good	Good	Good	Good	Good	Good	Good	Good
Fuel oil	Good	Good	Fair	Fair	Poor	Good	Good	Good
Kerosene	Good	Good	Fair	Gair	Poor	Good	Good	Good
Gasoline	Good	Good	Fair	Fair	Poor	Good	Good	Good
Petroleum-based hydraulic oil	Good	Good	Good	Good	Good	Good	Good	Good
Brake fluid	Poor	Fair	Poor	Poor	Poor	Fair	Fair	Good
Skydrol 500	Poor	Poor	Poor	Poor	Good	Poor	Good	Good
MIL-L-7808	Fair	Fair	Poor	Poor	Good	Good	Good	Good
MIL-L-23699	Fair	Fair	Poor	Poor	Good	Good	Good	Good
MIL-L-6082-A	Good	Good	Good	Good	Good	Good	Good	Good
MIL-L-5606	Good	Good	Good	Good	Poor	Good	Good	Good
MIL-L-2105	Fair	Fair	Good	Good	Poor	Fair	Fair	Good
MIL-G-10924	Good	Good	Good	Good	Poor	Good	Good	Good
Butane	Good	Good	Good	Good	Fair	Good	Good	Good
Ketone	Poor	Poor	Poor	Poor	Poor	Poor	Poor	Good
Ammonium gas, cold	Good	Good	Poor	Poor	Fair	Poor	Poor	Good
Fresh water	Good	Good	Poor	Poor	Good	Good	Good	Good
Salt water	Good	Good	Poor	Poor	Good	Good	Good	Good

Source: Chicago Rawhide

Table 2.5 Compound Compatibility Chart

	Lip material			
Chemical	N	P	S	F
Acetaldehyde	4	4	2	4
Acetamide 1	4	2	2	
Acetic acid, glacial	3	4	2	3
Acetic acid, 30%	2	4	1	2
Acetic anhydride	3	4	3	4
Acetone	4	4	3	4
Acetophenone	4	4	4	4
Acetyl chloride	4	4	3	1
Acetylene 1	4	2	1	
Acrylonitrile	4	4	4	3
Adipic acid	1	--	--	--
Alkazene (dibromoethylbenzene)	4	4	4	2
Alum-NH_3-Cr-K (aq)	1	4	1	4
Aluminum acetate (aq)	2	4	4	4
Aluminum chloride (aq)	1	1	2	1
Aluminum fluoride (aq)	1	--	2	1
Aluminum nitrate (aq)	1	--	2	1
Aluminum phosphate (aq)	1	--	1	1
Aluminum sulfate (aq)	1	4	1	1
Ammonia, anhydrous	2	4	3	4
Ammonia gas (Cold)	1	4	1	4
Ammonia gas (Hot)	4	4	1	4
Ammonium carbonate (aq)	4	4	--	--
Ammonium chloride (aq)	1	--	--	1
Ammonium hydroxide (conc)	4	4	1	2
Ammonium nitrate (aq)	1	2	--	--
Ammonium nitrite (aq)	1	--	2	--
Ammonium persulfate (aq)	4	4	--	--
Ammonium phosphate (aq)	1	--	1	--
Ammonium sulfate (aq)	1	4	--	4
Amyl acetate (Banana Oil)	4	4	4	4
Amyl alcohol	2	4	4	2
Amyl borate	1	--	--	1
Amyl chloronaphthalene	4	4	4	1
Amyl naphthalene	4	2	4	1
Anderol L-774 (MIL-L-7808D)	2	4	1	1
Aniline	4	4	4	3

Ratings: 1 = Minor effect, 2 = Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
Aniline dyes	4	4	3	2
Aniline hydrochloride	2	4	4	2
Animal fats	1	1	2	1
Ansul ether (anesthetics)	3	4	4	4
Aqua regia	4	4	4	2
Aroclor l248	3	4	2	1
Aroclor 1254	4	4	3	1
Aroclor 1260	1	4	2	1
Arsenic acid	1	3	1	1
Arsenic trichloride (aq)	1	--	--	--
Askarel	2	4	4	1
Asphalt	2	2	4	1
ASTM (Method D-471)				
#1	1	1	3	1
#2	1	1	3	1
#3	1	1	3	1
ASTM Reference Fuel A	See Mil-S-3136B Type I			
ASTM Reference Fuel B	See Mil-S-3136B Type III			
ASTM Reference Fuel C	2	4	4	1
ATF Type A	1	1	1	1
ATF Type I	1	1	2	1
ATF Type II	1	1	2	1
ATF Type F	1	1	2	1
ATF Type (Mercon)	1	1	2	1
Banana oil (amyl acetate)	4	4	4	4
Barium chloride (aq)	1	1	1	1
Barium hydroxide (aq)	1	4	1	1
Barium sulfate (aq)	1	4	1	1
Barium sulfide (aq)	1	4	1	1
Beer	1	4	1	1
Beer sugar liquors	1	4	1	1
Benzaldehyde	4	4	2	4
Benzene	4	4	4	1
Benzene sulfonic acid	4	4	4	1
Benzine (ligroin) (nitrobenzene) (petroleum ether)	1	1	4	1
Benzoic acid	3	3	3	1

Ratings: 1 = Minor effect, 2 = Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material N	P	S	F
Benzoyl chloride	4	4	--	1
Benzyl alcohol	4	4	2	1
Benzyl benzoate	4	4	--	1
Benzyl chloride	4	4	4	1
Biphenyl (diphenyl) (phenylbenzene)	4	4	4	1
Blast furnace gas	4	4	1	1
Bleach solutions	4	4	2	1
Borax	2	2	2	1
Bordeaux mixture	2	4	2	1
Boric acid 1	4	1	1	
Brake fluid	See Wagner 21B Brake Fluid			
B.P. Aero Hydraulic Fluid No. 1 (DTD585)	--	--	--	1
Brine	1	4	1	1
Bromine, anhydrous	4	4	4	1
Bromine trifluoride	4	4	4	4
Bromine water	4	4	4	1
Bromobenzene	4	4	4	1
Bunker oil	1	1	2	1
Butadiene	4	4	4	1
Butane	1	1	4	1
Butter (animal fat)	1	1	2	1
Butyl acetate	4	4	4	4
Butyl acetyl ricinoleate	3	--	--	1
Butadiene	4	4	4	1
Butane	1	1	4	1
Butter (animal fat)	1	1	2	1
Butyl acetate	4	4	4	4
Butyl acetyl ricinoleate	3	--	--	1
Butyl acrylate	4	4	--	4
Butyl alcohol	1	4	2	1
Butyl amine	3	4	4	4
Butyl benzoate	4	4	--	1
Butyl carbitol	4	4	4	1
Butyl Cellosolve	3	4	--	4
Butyl oleate	4	--	--	1
Butyl Stearak	2	--	--	1
Butylene 2	4	4	1	

Ratings: 1 = Minor effect, 2 = Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
Butyraldehyde	4	4	4	4
Calcium acetate (aq)	2	4	4	4
Calcium bisulfite (aq)	4	4	1	1
Calcium chloride (aq)	1	1	1	1
Calcium hydroxide (aq)	1	4	1	1
Calcium hypochlorite (aq)	2	4	2	1
Calcium nitrate (aq)	1	1	2	1
Calcium sulfide (aq)	1	4	2	1
Cane sugar liquors	1	4	1	1
Carbamate	3	4	--	1
Carbitol	2	4	2	2
Carbolic acid (Phenol)	4	4	4	1
Carbon bisulfide	3	3	4	1
Carbon dioxide	1	--	2	1
Carbonic acid	2	1	1	1
Carbon monoxide	1	--	1	1
Carbon tetrachloride	3	4	4	1
Castor oil	1	1	1	1
Cellosolve	4	4	4	3
Cellosolve acetate	4	4	4	4
Cellube 220	See MIL-L-19457			
Cellulube (Fryquel)	4	4	1	1
China wood oil (tung oil)	1	--	4	1
Chlorine (dry)	4	4	4	1
Chlorine (wet)	5	4	4	1
Chlorine dioxide	4	4	--	1
Chlorine trifluoride	4	4	4	4
Chloroacetic acid	4	4	--	4
Chloroacetone	4	4	4	4
Chlorobenzene	4	4	4	1
Chlorobromomethane	4	4	4	1
Chlorobutadiene	4	4	4	1
Chlorododecane	4	4	4	1
Chloroform	4	4	4	1
O-Chloronapthalene	4	4	4	1
1-Chloro-l-nitroethane	4	4	4	4
Chlorosulfonic acid	4	4	4	4

Ratings: 1 = Minor effect, 2 = Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
Chlorotoluene	4	4	4	1
Chrome plating solutions	4	4	2	1
Chromic acid	4	4	3	1
Citric acid	1	--	1	1
Clorox (sodium hypochlorite; NaOCl)	2	4	2	1
Coal tar (creosote)	1	1	4	1
Cobalt chloride (aq)	1	4	2	1
Coconut oil	1	1	1	1
Cod liver oil	1	1	2	1
Coke oven gas	4	4	2	1
Copper acetate (aq)	2	4	4	4
Copper chloride (aq)	1	1	1	1
Copper cyanide (aq)	1	1	1	1
Copper sulfate (aq)	1	4	1	1
Corn oil	1	1	1	1
Cottonseed oil	1	1	1	1
Creosote (coal tar)	1	1	4	1
Cresol	4	4	4	1
Cresylic acid	4	4	4	1
Cumene	4	4	4	1
Cyclohexane	1	1	4	1
Cyclohexanol	3	--	4	1
Cyclohexanone	4	4	4	4
p-Cymene	4	4	4	1
Decalin	4	--	4	1
Decane	1	1	2	1
Denatured alcohol	1	4	1	1
Detergent solutions	1	4	1	1
Developing fluids	1	--	1	1
Diacetone	4	4	4	4
Diacetone alcohol	4	4	2	4
Dibenzyl ether	4	--	--	4
Dibenzyl sebecate	4	4	3	2
Dibromoethylbenzene (alkazene)	4	4	4	2
Dibutylamine	4	4	3	4
Dibutyl ether	4	3	4	3
Dibutyl phthalate	4	4	2	3

Ratings: 1= Minor effect, 2 = Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

	Lip material			
Chemical	N	P	S	F
Dibutyl sebecate	4	4	2	2
o-Dichlorobenzene	4	4	4	1
Dichloro-isopropyl ether	4	3	4	3
Dicyclohexylamine	3	4	--	4
Diesel oil	1	1	4	1
Diethylamine	2	4	2	4
Diethylbenzene	4	--	4	1
Diethyl ether	4	3	4	4
Diethylene glycol	1	2	2	1
Diethyl sebecate	2	4	2	2
Diisobutylene	2	4	4	1
Diisopropylbenzene	4	--	--	1
Diisopropyl ketone	4	4	4	4
Diisopropylidene acetone (phorone)	4	4	4	4
Dimethyl aniline (xylidine)	3	4	4	4
Dimethyl ether (methyl ether; monomethyl ether)	1	4	1	1
Dimethyl formamide	2	4	2	4
Dimethyl phthalate	4	4	--	2
Dinitrotoluene	4	4	4	4
Dioctyl phthalate	3	4	3	2
Dioctyl sebecate	4	4	3	2
Dioxane	4	4	4	4
Dioxolane	4	4	4	4
Dipentene	2	4	4	1
Diphenyl (biphenyl; phenylbenzene)	4	4	4	1
Diphenyl oxides	4	4	3	1
Dowtherm oil	4	4	3	1
Dry cleaning fluids	3	4	4	1
Epichlorohydrin	4	4	4	4
Ethane	1	1	4	1
Engine oil (motor oil)	1	1	1	1
EP lubes	2	1	4	1
Ethanol (ethyl alcohol)	1	4	1	3
Ethanolamine	2	4	2	4
Ethyl acetate	4	4	2	4
Ethyl acetoacetate	4	4	2	4

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material N	P	S	F
Ethyl acrylate	4	4	2	4
Ethyl alcohol (ethanol)	1	4	1	3
Ethylbenzene	4	4	4	1
Ethylbenzoate	4	4	4	1
Ethyl Cellosolve	4	4	4	4
Ethyl cellulose	2	4	3	4
Ethyl chloride	1	4	4	1
Ethyl chlorocarbonate	4	4	4	1
Ethyl chloroformate	4	4	4	4
Ethyl ether	3	4	4	4
Ethyl formate	4	--	--	1
Ethyl mercaptan	4	--	3	2
Ethyl oxalate	4	4	4	1
Ethyl pentachlorobenzene	4	4	4	1
Ethyl silicate	1	--	--	1
Ethylene	1	--	--	1
Ethylene chloride	4	4	4	2
Ethylene chlorohydrin	4	4	3	1
Ethylenediamine	1	4	1	4
Ethylene dichloride	4	4	4	1
Ethylene glycol	1	3	1	1
Ethylene oxide	4	4	4	4
Ethylene trichloride	4	4	4	1
Fatty acids	2	--	3	1
Ferric chloride (aq)	1	1	2	1
Ferric nitrate (aq)	1	1	3	1
Ferric sulfate (aq)	1	1	2	1
Fish oil	1	--	1	1
Fluorinated cyclic ethers	--	--	--	--
Fluorine (liquid)	4	4	4	2
Fluorobenzene	4	4	4	1
Fluoroboric acid	1	--	--	--
Fluorocarbon oils	--	--	--	--
Fluorolube	1	--	1	2
Fluorosilicic acid (hydrofluosilicic Acid)	1	--	4	1
Formaldehyde (RT)	3	4	2	4
Formaldehyde, 37%	3	4	2	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
Formic acid	2	--	2	3
Freon 11	2	--	4	1
Freon 12	1	1	4	2
Freon 13	1	--	4	1
Freon 14	1	--	4	1
Freon 21	1	--	4	4
Freon 22	4	2	4	4
Freon 31	4	--	--	4
Freon 32	1	--	--	4
Freon 112	3	--	4	1
Freon 113	1	--	4	2
Freon 114	1	--	4	2
Freon 115	1	--	--	2
Freon 142b	1	--	--	4
Freon 152a	1	--	--	4
Freon 218	1	--	--	1
Freon C316	1	--	--	--
Freon C318	1	--	--	2
Freon 13B1	1	--	4	1
Freon 114B2	2	--	4	2
Freon 502	2	--	--	2
Freon TF	1	--	4	2
Freon T-WD602	2	--	4	1
Freon TMC	2	--	3	1
Freon T-P35	1	--	1	1
Freon TA	1	--	3	3
Freon TC	1	--	4	1
Freon MF	1	--	4	2
Freon BF	2	--	4	1
Fuel oil	1	1	4	1
Fumaric acid	1	4	2	1
Furan, furfuran	4	4	--	--
Furfural	4	4	4	4
Fyrquel (Cellulube)	4	4	1	1
Gallic acid2		4	--	1
Gasohol (10% ethanol or methanol)	2	4	4	3
Gasoline	2	4	4	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Compound Compatibility Chart

	Lip material			
Chemical	N	P	S	F
Gelatin	1	4	1	1
Gauber’s salt (aq)	4	4	--	1
Glucose	1	--	1	1
Glue	1	--	1	1
Glycerin	1	3	1	1
Glycols	1	4	1	1
Grease	1	2	1	1
Green sulfate liquor	2	2	1	1
Halowax oil	4	--	4	1
N-Hexaldehyde	4	--	2	4
Hexane	1	1	4	1
N-Hexene-1	2	1	4	1
Hexyl alcohol	1	4	2	1
Hydrazine	2	--	3	4
Hydraulic oil (petroleum)	1	1	3	1
Hydrobromic acid	4	4	4	1
Hydrobromic acid 40%	4	4	4	1
Hydrochloric acid (cold) 37%	3	4	3	1
Hydrochloric acid (hot) 37%	4	4	4	2
Hydrocyanic acid	2	4	3	1
Hydrofluoric acid (conc.) cold	4	4	4	1
Hydrofluoric acid (conc.) hot	4	4	4	3
Hydrofluoric acid, anhydrous	4	4	4	4
Hydrofluosilicic acid (fluosilicic acid)	1	--	4	1
Hydrogen gas	1	2	3	1
Hydrogen peroxide (90%)	4	4	2	2
Hydrogen sulfide (wet) cold	4	4	3	4
Hydrogen sulfide (wet) hot	4	4	3	4
Hydroquinone	3	4	--	2
Hypochlorous acid	4	4	--	1
Iodine pentafluoride	4	4	4	4
Iodoform	--	--	--	--
Isobutyl alcohol	2	4	1	1
Isooctane	1	1	4	1
Isophorone	4	4	4	4
Isopropyl acetate	4	4	4	4
Isopropyl alcohol	2	4	1	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Compound Compatibility Chart

	Lip material			
Chemical	N	P	S	F
Isopropyl chloride	4	4	4	1
Isopropyl ether	2	3	4	4
JP3, JP4, JP5	See MIL-J-5624G			
JP6	See MIL-F-25656 B			
Kerosene	1	1	4	1
Lacquers	4	4	4	4
Lacquer solvents	4	4	4	4
Lactic acid (cold)	1	4	1	1
Lactic acid (hot)	4	4	2	1
Lard	1	1	2	1
Lavender oil	2	2	4	1
Lead acetate (aq)	2	4	4	4
Lead nitrate (aq)	1	--	2	--
Lead sulfamate (aq)	2	4	2	1
Legroin (benzine) (nitrobenzine) (pet ether)	1	1	4	1
Lime bleach	1	4	2	1
Lime sulfur	4	4	1	1
Lindol (hydraulic fluid)	4	4	3	2
Linoleic acid	2	--	2	2
Linseed oil	1	1	1	1
Liquefied petroleum gas	1	3	3	1
Lubricating oils (petroleum)	1	1	4	1
Lye	2	4	2	2
Magnesium chloride (aq)	1	--	1	1
Magnesium hydroxide (aq)	2	4	--	1
Magnesium sulfate (aq)	1	4	1	1
Maleic acid	4	4	--	1
Maleic anhydride	4	4	--	4
Malic acid	1	4	2	1
Mercury chloride (aq)	1	--	--	1
Mercury	1	--	--	1
Mesityl oxide	4	4	4	4
Methane	1	1	4	2
Methanol (methyl alcohol)	1	4	1	4
Methyl acetate	4	4	4	4
Methyl acrylate	4	4	4	4
Methylacrylic acid	4	4	4	4

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Compound Compatibility Chart

Chemical	Lip material N	P	S	F
Meethyl alcohol (methanol)	1	4	1	4
Methyl bromide	2	3	--	1
Methyl butyl ketone (propyl acetone)	4	4	3	4
Methyl Cellosolve	3	4	4	4
Methyl chloride	4	4	4	2
Methyl cyclopentane	4	4	4	2
Methylene chloride	4	4	4	2
Methyl ether (dimethyl ether; monomethyl ether)	1	4	1	1
Methyl ethyl ketone	4	4	4	4
Methyl formate	4	--	--	--
Methyl isobutyl ketone	4	4	4	4
Methyl methacrylate	4	4	4	4
Methyl oleate	4	--	--	2
Methyl salicylate	4	--	--	--
Milk	1	4	1	1
Mineral oil	1	1	2	1
Mineral spirits	2	--	4	1
Monochlorobenzene	4	4	4	1
Monomethyl aniline	4	4	--	2
Monoethanol amine	4	4	2	4
Monomethyl ether (methyl ether; dimethyl ether)	1	4	1	1
Monovinyl acetylene	1	--	2	1
Mustard gas	--	--	1	--
Naphtha	2	2	4	1
Naphthalene	4	--	4	1
Naphthalenic acid	2	--	4	1
Natural gas	1	2	1	1
Neat's-foot oil	1	1	2	1
Neville acid	4	4	4	1
Nickel acetate (aq)	2	4	4	4
Nickel chloride	1	3	1	1
Nickel sulfate (aq)	1	4	1	1
Niter cake	1	4	1	1
Nitric acid (conc)	4	4	4	3
Nitric acid (dilute)	4	4	2	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Compound Compatibility Chart

	Lip material			
Chemical	N	P	S	F
Nitric acid, red puming	4	4	4	4
Nitrobenzene	4	4	4	2
Nitrobenzene (petroleum ether)	1	1	4	1
Nitroethane	4	4	4	4
Nitrogen	1	1	1	1
Nitrogen tetroxide	4	4	4	4
Nitromethane	4	4	4	4
Octachlorotoluene	4	4	4	1
Octadecane	1	2	4	1
N-Octane	2	4	4	1
Octyl alcohol	2	4	2	1
Oleic acid	3	4	4	2
Oleum spirits	2	--	4	1
Olive oil	1	1	3	1
O-Dichlorobenzene	4	4	4	1
Oronite 8200		See MIL-H-8446B		
Oxalic acid	2	--	2	1
Oxygen, cold	2	2	1	1
Oxygen (200-400˚F)	4	4	2	2
Ozone	4	2	1	1
Paint thinner, Duco	4	4	4	2
Palmitic acid	1	--	4	1
Peanut oil 1		1	1	1
Perchloric acid	4	4	4	1
Perchloroethylene	2	4	4	1
Petroleum, below 250˚F	1	2	2	1
Petroleum, above 250˚F	4	4	4	2
Phenol (carbolic acid)	4	4	4	1
Phenylbenzene		See Diphenyl		
Phenyl ethyl ether	4	4	4	4
Phenyl hydrazine	4	4	--	1
Phorone (diisopropylidene acetone)	4	4	4	4
Phosphate ester	4	4	1	1
Phosphoric acid, 20%	2	--	2	1
Phosphoric acid, 45%	4	--	3	1
Phosphorus trichloride	4	--	--	1
Pickling solution	4	4	4	2

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
Picric acid	2	--	4	1
Pinene	2	4	4	1
Pine oil	4	--	4	1
Piperidine	4	4	4	4
Plating solution, chrome	--	--	4	1
Plating solution, others	1	--	4	1
Polyvinyl acetate emulsion	--	--	--	--
Potassium acetate (aq)	2	4	4	4
Potassium chloride (aq)	1	1	1	1
Potassium cupro cyanide (aq)	1	1	1	1
Potassium cyanide (aq)	1	1	1	1
Potassium dichromate (aq)	1	1	1	1
Potassium hydroxide (aq)	2	4	3	4
Potassium nitrate (aq)	1	1	1	1
Potassium sulfate (aq)	1	4	1	1
Producer gas	1	2	2	1
Propane	1	1	4	1
i-Propyl acetate	4	4	4	4
n-Propyl acetate	4	4	4	4
Propyl acetone (methyl butyl ketone)	4	4	3	4
Propyl alcohol	1	4	1	1
Propyl nitrate	4	4	4	4
Propylene	4	4	4	1
Propylene oxide	4	4	4	4
Pydraul, 10E, 29 ELT	4	4	4	1
Pydraul, 30E, 50E, 65E, 90E	4	4	1	1
Pydraul, 115E	4	4	4	1
Pydraul, 230E, 312C, 540C	4	4	4	1
Pyranol, transformer oil	1	1	4	1
Pyridine	4	4	4	4
Pyroligneous acid	4	4	--	4
Pyrrole	4	4	2	4
Radiation	3	3	3	4
Rapeseed oil	2	2	4	1
Red oil (MIL-H-5606)	1	1	4	1
RJ-1 (MIL-F-25558 B)	1	1	4	1
RP-1 (MIL-F-25576 C)	1	1	4	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

	Lip material			
Chemical	N	P	S	F
SAE 30	1	1	1	1
SAE 90	1	1	4	1
Sal ammoniac	1	1	2	1
Salicylic acid	2	--	--	1
Salt water	1	4	1	1
Sewage	1	4	2	1
Silicate esters	2	--	4	1
Silicone greases	1	1	3	1
Silicone oils	1	1	3	1
Silver nitrate	2	1	1	1
Skydrol 500	4	4	3	4
Skydrol 7000	4	4	3	2
Soap solutions	1	4	1	1
Soda ash	1	--	1	1
Sodium acetate (aq)	2	4	4	4
Sodium bicarbonate (aq) (baking soda)	1	--	1	1
Sodium bisulfite (aq)	1	4	1	1
Sodium borate (aq)	1	--	1	1
Sodium chloride (aq)	1	--	1	1
Sodium cyanide (aq)	1	--	1	1
Sodium hydroxide (aq)	2	3	2	2
Sodium hypochlorite (aq) (Chlorox)	2	4	2	1
Sodium metaphosphate (aq)	1	--	--	1
Sodium nitrate (aq)	2	--	4	--
Sodium perborate (aq)	2	--	2	1
Sodium peroxide (aq)	2	4	4	1
Sodium phosphate (aq)	1	1	4	1
Sodium silicate (aq)	1	--	--	1
Sodium sulfate (aq)	1	4	1	1
Sodium thiosulfate (aq)	2	4	1	1
Soybean oil	1	1	1	1
Stannic chloride (aq)	1	--	2	1
Stannous chloride (aq)	1	--	2	1
Steam under 300˚F	4	4	3	4
Steam over 300˚F	4	4	4	4
Stearic acid	2	--	2	--
Stoddard solvent	1	1	4	1

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N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

	Lip material			
Chemical	N	P	S	F
Styrene	4	4	4	2
Sucrose solution	1	4	1	1
Sulfite liquors	2	4	4	1
Sulfur	4	4	3	1
Sulfur chloride (aq)	3	4	3	1
Sulfur dioxide (dry)	4	4	2	1
Sulfur dioxide (wet)	4	4	2	1
Sulfur dioxide (liquefied under pressure)	4	4	2	1
Sulfur hexafluoride	2	4	2	1
Sulfur trioxide	4	4	2	1
Sulfuric acid (dilute)	3	2	4	1
Sulfuric acid (conc)	4	4	4	1
Sulfuric acid (20% oleum)	4	4	4	1
Sulfurous acid	2	4	4	1
Tannic acid	1	4	2	1
Tar, bituminous	2	4	2	1
Tartaric acid	1	--	1	1
Terpineol	2	--	--	1
Tertiary-Butyl alcohol	2	4	2	1
Tertiary-Butyl catechol	4	4	--	1
Tertiary-Butyl mercaptan	4	4	4	1
Tetrabromoethane	4	4	4	1
Tetrabromomethane	4	--	4	1
Tetrabutyl titanate	2	--	--	1
Tetrochloroethylene	4	4	4	1
Tetraethyllead	2	--	--	1
Tetrahydrofuran	4	4	4	4
Tetralin	4	--	4	1
Thionyl chloride	4	4	--	2
MIL-L-644 B	1	2	3	--
MIL-L-2104 B	1	1	3	1
MIL-L-2105 B	1	1	3	1
MIL-G-2108	1	1	3	1
MIL-S-3136 B Type I	1	2	4	1
MIL-S-3136 B Type II	1	--	4	1
MIL-S-3136 B Type III	1	--	4	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
MIL-S-3136 B Type IV	1	1	3	1
MIL-S-3136 B Type V	1	1	3	1
MIL-S-3136 B Type VI	1	1	3	1
MIL-S-3136 B Type VII	1	--	4	1
MIL-L-3150 A	1	1	3	1
MIL-L-3503	1	1	3	1
MIL-L-3545 B	2	2	4	1
MIL-C-4339 C	1	1	3	1
MIL-G-4343 B	2	1	4	1
MIL-L-5020 A	1	2	4	1
MIL-J-5161 F	1	--	4	1
MIL-C-5545 A	2	2	4	1
MIL-H-5559 A	1	3	2	2
MIL-F-5566	1	--	1	1
MIL-F-5602	1	1	3	1
MIL-H-5606 B (Red Oil)	1	1	4	1
MIL-J-5624 G JP-3, JP-4, JP-5	1	2	4	1
MIL-O-6081 C	1	1	3	1
MIL-L-6082 A	1	1	1	1
MIL-L-6082 C	1	1	3	1
MIL-H-6083 C	1	1	3	1
MIL-L-6085 A	1	3	3	1
MIL-L-6086 B	1	1	3	1
MIL-L-6387 A	1	--	3	1
MIL-C-6529 C	2	2	4	1
MIL-F-7024 A	1	2	4	1
MIL-H-7083 A	1	3	2	2
MIL-G-7118 A	1	3	3	1
MIL-G-7187	1	1	3	1
MIL-G-7421 A	1	--	3	1
MIL-H-7644	2	2	4	1
MIL-L-7645	2	2	4	1
MIL-G-7711 A	1	1	3	1
MIL-L-7808 D	2	4	1	1
MIL-L-7808 E	2	4	1	2
MIL-L-7808 F	1	3	3	1

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N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
MIL-L-7870 A	1	1	3	1
MIL-C-8188 C	1	3	3	1
MIL-A-8243 B	1	3	2	2
MIL-L-8383 B	1	1	3	1
MIL-H-8446 B (MLO-8515)	2	3	4	1
MIL-1-8660 B	1	--	4	1
MIL-L-9000 F	1	2	4	1
MIL-T-9188 B	4	4	4	4
MIL-L-9236 B	1	3	4	1
MIL-L-10295 A	1	1	3	1
MIL-L-10324 A	1	1	3	1
MIL-G-10924 B	1	1	3	1
MIL-L-11734 B	1	3	3	1
MIL-O-11773	1	3	3	1
MIL-P-12098	2	2	4	1
MIL-H-13862	1	1	3	1
MIL-H-13866 A	1	1	3	1
MIL-H-13910 B	2	2	4	1
MIL-H-13919 A	1	1	3	1
MIL-L-14107 B	3	--	4	1
MIL-L-15017	1	1	3	1
MIL-L-15018 B	1	1	3	1
MIL-L-15019 C	1	1	3	1
MIL-L-15719 A	2	2	4	1
MIL-G-15793	1	3	3	1
Titanium tetrachloride	2	4	4	1
Toluene	4	4	4	1
Toluene diisocyanate	4	4	4	4
Transformer oil	1	2	2	1
Transmission fluid Type A	1	1	2	1
Triacetin	2	4	--	4
Triaryl phosphate	4	4	3	1
Tributoxy ethyl phosphate	4	4	--	1
Tributyl mercaptan	4	4	4	1
Tributyl phosphate	4	4	4	4
Trichloroacetic acid	2	4	--	3

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
Trichloroethane	4	4	4	1
Trichloroethylene	4	4	4	1
Tricresyl phosphate	4	4	3	1
Triethanolamine	2	4	--	4
Triethylaluminum	4	4	--	2
Triethyl borane	4	4	--	1
Trinitrotoluene	4	4	--	2
Trioctyl phosphate	4	4	3	2
Tung oil (china wood oil)	1	--	4	1
Turbine oil	2	1	4	1
Turpentine	1	2	4	1
Unsymmetrical dimethyl hydrazine (UDMH)	2	--	4	4
Varnish	2	4	4	1
Vegetable oils	1	1	2	1
Versilube F-50	1	1	3	1
Vinegar	2	4	1	1
Vinyl chloride	4	4	--	1
Wagner 21 B brake fluid	3	--	3	4
Water	1	4	1	1
Whiskey, wines	1	4	1	1
White pine oil	2	--	4	1
White oil	1	1	4	1
Wood oil	1	1	4	1
Xylene	4	4	4	1
Xylidine (dimethyl aniline)	3	4	4	4
Zeolites	1	--	--	1
Zinc acetate (aq)	2	4	4	4
Zinc chloride (aq)	1	4	1	1
Zinc sulfate (aq)	1	4	1	1
TT-T656b	4	4	4	4
VV-B-680	2	2	4	1
VV-G-632	1	1	3	1
VV-G-671c	1	1	3	1
VV-H-910	2	2	4	1
VV-I-530a	1	1	3	1
VV-K-211d	1	--	4	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material N	P	S	F
VV-K-220a	1	2	4	1
VV-L-751b	2	2	4	1
VV-L-800	1	1	3	1
VV-L-820b	1	1	3	1
VV-L-825a Type I	1	1	3	1
VV-L-825a Type II	1	1	3	1
VV-L-825A Type III	2	2	4	1
VV-O-526	1	1	3	1
VV-P-216A	1	1	3	1
VV-P-236	2	2	4	1
51-F-23	1	1	3	1
MIL-F-16929 A	1	3	3	1
MIL-L-16958 A	1	1	3	1
MIL-F-17111	1	1	3	1
MIL-L-17331 D	1	1	3	1
MIL-L-17353 A	1	--	3	1
MIL-L-17672 B	1	1	3	1
MIL-L-18486 A	1	1	3	1
MIL-G-18709 A	1	1	3	1
MIL-H-19457 B	4	4	3	3
MIL-F-19605	1	--	4	1
MIL-L-19701	1	3	3	1
MIL-L-21260	1	1	3	1
MIL-S-21568 A	1	1	4	1
MIL-H-22072	1	3	2	2
MIL-L-22396	1	1	3	1
MIL-L-23699 A	1	3	3	1
MIL-G-23827 A	1	3	3	1
MIL-G-25013 D	1	2	4	1
MIL-F-25172	1	--	4	1
MIL-L-25336 B	1	3	3	1
MIL-F-25524 A	1	--	4	1
MIL-G-25537 A	1	1	3	1
MIL-F-25558 B (RJ-1)	1	1	3	1
MIL-F-25576 C (RP-1)	1	1	4	1
MIL-H-25598	1	1	3	1
MIL-F-25656 B	1	--	4	1

Ratings: 1 = Minor effect, 2= Moderate effect, 3 = Static only, 4 = Not recommended, -- = Insufficient data.

N= Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Table 2.5 Continued

Chemical	Lip material			
	N	P	S	F
MIL-L-25681 C	1	2	4	1
MIL-G-25760 A	1	3	4	1
MIL-L-25968	1	3	3	1
MIL-L-26087 A	1	1	3	1
MIL-G-27343	1	--	4	1
MIL-H-27601 A	2	2	4	1
MIL-G-27617	4	--	4	1
MIL-I-27686 D	1	3	2	2
MIL-L-27694 A	1	--	4	1
MIL-L-46000 A	1	3	3	1
MIL-H-46001 A	1	1	3	1
MIL-L-46002	1	--	3	1
MIL-H-46004	1	1	3	1
MIL-P-46046 A	2	2	4	1
MIL-H-81019 B	1	1	3	1
MIL-S-81087	1	--	4	1
MIL-G-81322	2	--	--	--
O-A-548 a	1	3	2	2
O-T-634 b	3	4	4	1
P-S-661 b	1	--	4	1
P-D-680	1	--	4	1
TT-N-95 a	1	--	4	1
TT-N-97 B	1	3	4	1
TT-I-735 b	1	--	1	1
TT-S-735 Type I	1	2	4	1
TT-S-735 Type II	1	--	4	1
TT-S-735 Type III	1	--	4	1
TT-S-735 Type IV	1	1	3	1
TT-S-735 Type V	1	1	3	1
TT-S-735 Type VI	1	1	3	1
TT-S-735 Type VII	1	--	4	1

Ratings: 1 = Minor effect, 2 = Moderate effect, 3 = Static only, 4 = Not recommended, - = Insufficient data.

N = Nitrile; P = polyacylate; S = silicone; F = fluoroelastomer.

Source: Chicago Rawhide

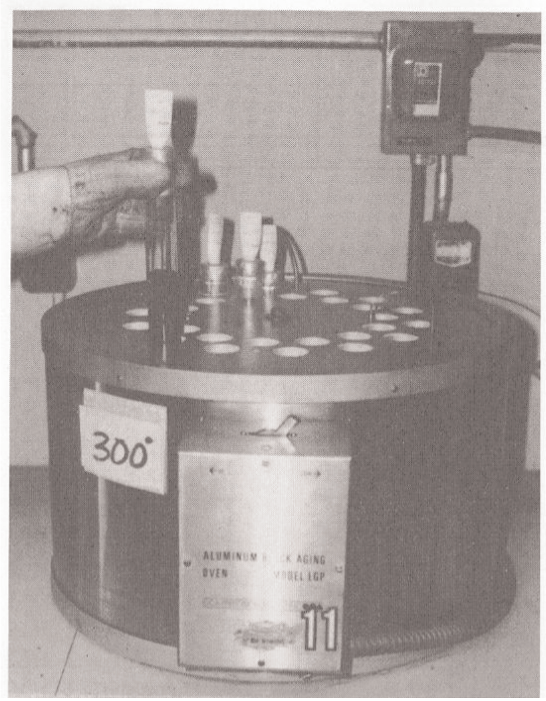


Fig. 2.18 Aluminum block for aging elastomers in fluids.

The presence of oxygen can increase the rate of deterioration for some compounds and decrease it for others. Many dynamic applications result in a churning of the oil that aerates the oil and exposes the elastomer to a continuous supply of oxygen. The sealed test tube will not duplicate the application, and a procedure has been developed that bubbles a known quantity of dry air into the test tube [32, 33].

It is recommended that the immersion tests use the fluid that the seal material will be exposed to in the application. If it is impractical to use an application oil, then ASTM has defined several standard test fluids. There is a low-swell oil [ASTM #1--aniline point 255˚F (124˚C)], a medium-swell oil [ASTM #2 oil--aniline point 200˚F (93˚C)], and a high-swell oil [ASTM #3 oil--aniline point 158˚F (70˚C)]. If the elastomer is exposed to fuels, then reference fuel A (100% isooctane) is representative of a low-swelling gasoline, and reference fuels B (70% isooctane, 30% toluene) and C (50% isooctane, 50% toluene) simulate high-swell gasolines.

Immersion tests are not intended to predict the service performance of elastomers. They can be used to develop comparative data in specific fluids to screen out those elastomers that have obvious deficiencies. Bench and field tests of the seals in the actual application hardware give the best measure of how a seal will work in the field.

2.3.13 Standard ASTM Classification System for Elastomeric Materials

A line call-out system is described in ASTM D2000 and has been developed to describe the ASTM test requirements (Table 2.2) and specifications that an elastomer must meet for a given application. The standard dimensions for the system described in D2000 are in SI units, and the line call-out begins with M if metric units are employed. If English units are used, then no letter will appear. The next position of the line call-out indicates the grade of the material. A 1 indicates that basic minimum test requirements are all that is required. If a 2, 3. . . or 8 appears, then additional requirements beyond the minimum will be specified.

The basic minimum test requirements are defined by the next two letters (Table 2.6). The first letter ranges from A through J and specifies the test temperature in degrees Celsius. The material will pass the heat requirements if the change in tensile strength is within ±30% of the initial values after heat aging at the test temperature for 70 hr. The elongation decrease must be less than 50%, and the material hardness must be within ±15 points of the original values. The second letter (A to K) determines the resistance

Table 2.6 Basic Requirements for Heat Resistance and Volume Swell in ASTM Oil #3 for Vulcanized Elastomers

First letters of call-out (establishes test temperature)		Second letter of call-out (establishes max volume swell in ASTM #3 at test temperature)	
Letter	Test Temp. (˚C)	Letter	Maximum volume swell (%), 70 hr soak
A	70	A	No requirement
B	100	B	140
C	125	C	120
D	150	D	100
E	175	E	80
F	200	F	60
G	225	G	40
H	250	H	30
J	275	K	10

of the elastomer to swell after soaking in a standard test fluid (ASTM Oil #3) for 70 hr at the test temperature specified by the first letter.

These letters are followed by a three-digit number to specify the initial hardness and tensile strength for the material. The first number indicates the material durometer hardness (6 for 60 ± 5, 7 for 70 ± 5, etc.), and the second and third indicate the minimum tensile strength (10 for 10 MPa, 20 for 20 MPa, etc.)

An example of the simplest line call-out is M1 DH 815. The M indicates that the test results are reported in SI units. The 1 means that only the basic test requirements need be met. D indicates that the test temperature is 150˚C, and the H signifies that the maximum volume swell of the material in ASTM #3 oil after a 70 hr soak at 125˚C is 30%. The letter D also means the elastomer will be heat-aged in air at 150˚C for 70 hr. After heat aging, the elongation loss will be less than 50%, the hardness change will lie within ±15 points of the original values, and the tensile strength change will be within ±30%. The 8 indicates that the material's initial durometer specification is 80 ± 5, and the 15 means that the tensile strength must be greater than 15 MPa.

If a 2 to 8 appears as the first digit in the line call-out, then additional tests are specified. The additional tests are defined by a suffix letter (S) followed by two numbers. The first number defines the ASTM test method (Table 2.7), and the second number defines the test temperature (Table 2.8).

As an example, consider the line call-out

M 5 DH 815 A26 C12 E016 E036 F27

In addition to the basic requirements already discussed for the DH 815 call-out, five additional test requirements have been added. The tests and specifications can be determined from the codes with the aid of Tables 2.7 and 2.8.

A26 is a heat resistance test using method ASTM D865 for 70 hr at a temperature of 150˚C.

C12 is an ozone resistance test using ASTM test method D1171 (ozone "A") at 38˚C.

E016 is an oil resistance test using ASTM test method D471 in ASTM Oil #1 at 150˚C.

E036 is an oil resistance test using ASTM test method D471 in ASTM Oil #3 at 150˚C.

F27 is a low-temperature test using ASTM test method D1053 at -40˚C.

Table 2.7 Suffix Letters and Test Methods for Vulcanized Elastomers

Suffix letter	Test description	First suffix number (ASTM test method)							
		1	2	3	4	5	6	7	8
A	Heat resistance –	D573 –70 hr	D865 –70 hr	D865 168 hr	D573 168 hr				
B	Compression set	D395 22 hr, solid-	D395 70 -hr, solid-	D395 22 hr, --piled-	D395 20 hr, - piled				
C	Ozone/weather resistance	D1171 ozone "A"	D1171 weather	D1171 ozone "B"	–	–	–	–	–
D	Compression-deflection resistance	D575 "A"	D575 "B"	–	–	–	–	–	–
EO	Fluid resistance (oils and lubricants)	D471 oil #l, 70 hr	D471 oil #2, 70 hr	D471 oil #3, 70 hr	D471 oil #1, 168 hr	D471 oil #2, `` 168 hr	D471 oil #3, 168 hr	D471 fluid 101, 70hr	D471 designated fluid, 70 hr
EF	Fluid resistance (fuels)	D471 Fuel A, 70 hr	D471 Fuel B, 70 hr	D471 Fuel C, 70 hr	D471 Fuel D, 70 hr	D471 gasohol, 70 hr	–	–	–
EA	Fluid resistance (aqueous)	D471 distilled water, 70 hr	D471 - water/glycol, 70 hr	–	–	–	–	–	
F	Low-temperature	D2137 "A"-	D1053 22 hr	D2137	D1329 resistance	D1329 9.3.2, 3 min	–	– A-9.3.2,	–
G	Tear resistance	D624 Die B	D624 Die C	–	–	–	–	–	–
H	Flex fatigue resistance	D430 A	D430 B	D430 C	–	–	–	–	–
J	Abrasion resistance[a]								
K	Adhesion	D429 A	D429 B	–	–	–	–	–	–
M	Flammability resistance[a]								
N	Impact resistancea								
P	Staining resistance	D429 A	D429 B	–	–	–	–	–	–
R	Resilience	D945	–	–	–	–	–	–	–
Z	Special; specified by user/supplier[a]								

[a] To be specified.

Table 2.8 Numbers to Indicate Test Temperatures for Vulcanized Elastomers

Applicable suffix letters	Second suffix number	Test temp. (˚C)
A, B, C, EA, EF, EO, G, K	11	275
	10	250
	9	225
	8	200
	7	175
	6	150
	5	125
	4	100
	3	70
	2	38
	1	23
	0	Ambient
F	1	23
	2	0
	3	-10
	4	-18
	5	-25
	6	-35
	7	-40
	8	-50
	9	-55
	10	-65
	11	-75
	12	-80

3
Lubricants

3.1 LUBRICATION THEORY

The most common use of a radial lip seal is to prevent lubricants from leaking out of a cavity or sump that contains gears or some other mechanical device that requires lubrication to prevent excessive wear. The principal modes of lubrication are thick film or hydrodynamic lubrication, mixed lubrication, boundary lubrication, elastohydrodynamic lubrication, and extreme pressure (EP) lubrication. The unexpected transition from one regime to another can result in catastrophic machine failure if the application lubricant is not properly formulated. Lubricants are formulated to minimize wear of moving mechanical components, and in many cases little thought is given to the interaction of the lubricant and the seal material. The radial lip seal is a friction device and depends upon the application lubricant to minimize the seal lip and shaft wear. The seal user and designer must have a basic understanding of lubricants and lubricant theory to ensure that the best seal is selected for the application [8, 116, 186].

Friction is the resistance to motion between two surfaces. It can be expressed as the force required to keep one surface in motion relative to another. The coefficient of friction (μ) is a dimensionless parameter that is the ratio of this frictional force to the applied load (P) between the two surfaces. Lubricants are used to reduce the coefficient of friction and thus minimize the frictional losses. Lubricants also have internal frictional

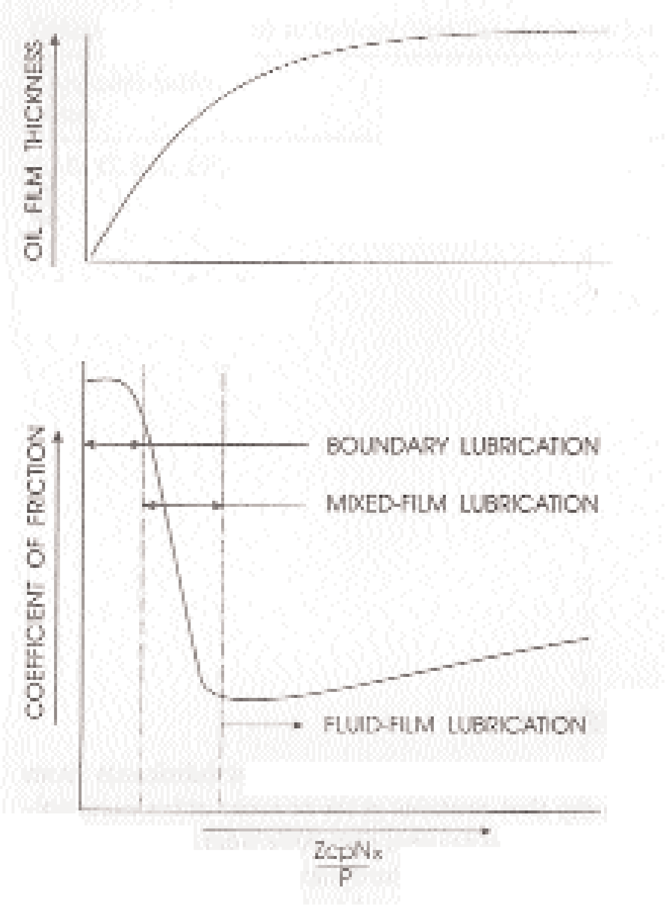


Fig. 3.1 Stribeck curve to define lubrication regimes.

forces that resist flow. The resistance to fluid flow is known as viscosity, (Zcp). The seal engineer can study how well equipment is lubricated by measuring and plotting the coefficient of friction μ versus the dimensionless parameter G, which is equal to (Zcp NS/P). The curve generated is known as a Stribeck curve (Fig. 3.1).

The coefficient of friction is independent of the parameter G in the boundary lubrication regime. The applied load is carried by microscopic peaks on the surfaces called asperities, and the fluid film between the surfaces is approximately the same thickness as the average asperity height (Fig. 3.2a). The lubricant must be formulated to create a film on the asperities that cannot be easily dislodged and will prevent component wear.

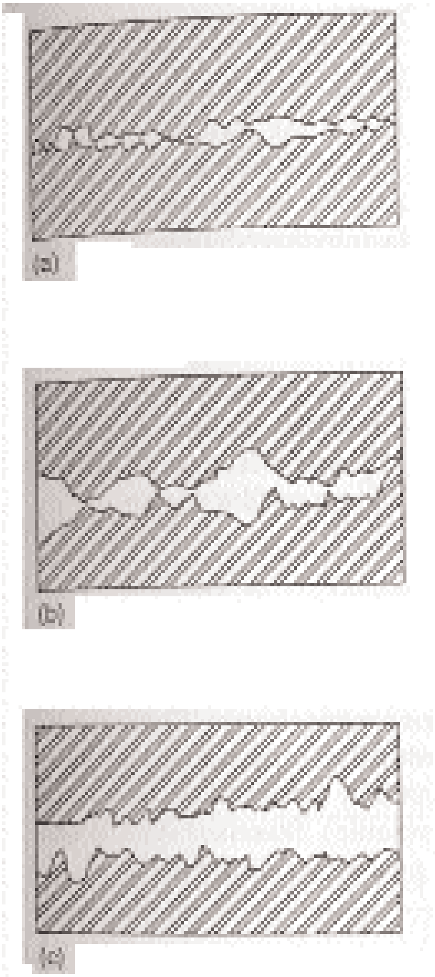


Fig. 3.2 Lubrication regimes. (a) Boundary lubrication, where performance is essentially dependent on the boundary film; (b) mixed film lubrication, where both the bulk lubrication and boundary film play a role; (c) fluid film lubrication, where surfaces are separated by bulk lubrication film.

Most machinery operates in the mixed film lubrication regime (Fig. 3.2b). The fluid film becomes thicker and the coefficient of friction drops as the parameter G becomes larger. The load is supported partially by the fluid film and partially by the surface asperities.

Full fluid film lubrication is rarely attained with most machine operations. The surfaces are fully separated and supported by the film (Fig. 3.2c) As the parameter G increases, the coefficient of friction will pass through a minimum and then increase linearly. Full fluid film lubrication is associated with high-speed, low-load applications that are lubricated with low-viscosity fluids.

The pressure between meshing gears can become very large as the fluid film is pinched. This rapid pressure increase causes the local viscosity to increase rapidly, and the film becomes “stiff.” The stiff fluid film becomes harder than the gear teeth surfaces and causes local deformation of the metal. This form of lubrication is called elastohydrodynamic (EHD), which means deformation without permanent distortion. Special additives for extreme pressure (EP) are required to prevent damage to the gear teeth.

Oils with different additive packages are formulated for different applications and lubrication regimes. This chapter describes various lubricant types and how these lubricants influence radial lip seal selection. It should be noted that the oil seal typically operates in the mixed film region.

3.2 BASE OIL

The base oil and additives determine the properties of the lubricant [203]. The selection of base oil is extremely important because it will determine the fundamental lubricant properties that additives cannot change. The base oil properties are dependent upon the type of crude oil selected and the process used to refine it. Crude and base oils are typically classified as paraffinic (waxy), naphthenic (nonwaxy), or aromatic (volatile). Standard ASTM test methods exist to measure crude and base oil properties (Table 3.1).

3.2.1 Aniline Point (ASTM D611)

The aniline point of an oil is the temperature at which a 50:50 mixture of the oil and freshly distilled aniline form a single phase. It may be reported in either Celsius or Fahrenheit degrees. Since aniline is a pure aromatic (volatile) compound, the aniline point is a measure of solubility for aromatic materials and thus a measure of the aromaticity of an oil. It is sometimes used to determine how well various additives will be absorbed by the base oil. It is dependent upon the chemical composition of the oil and its molecular weight (or viscosity). The more paraffinic an oil, the higher its aniline point; the higher the viscosity, the higher the aniline point. Typical aniline point ranges by oil type are given in Table 3.2.

3.2.2 Color (ASTM D1500)

There are a number of color scales used in the oil industry, but the scale described in ASTM D1500 is the most widely used. Like most other tests, ASTM D1500 describes a comparative test in which the color of an

Table 3.1 Measurement of Crude and Base Oil Properties

Aniline point	Temperature at which a 50:50 mixture of oil and aniline form a single phase	D611
Color		D1500
Flash point	Temperature at which open flame will ignite oil vapors	D92, D93
API gravity	$\frac{141.5}{\text{specific gravity}} -131.5$	D287,D1298
Viscosity gravity constant (VGC)	Formula calculation; removes viscosity variable	D2501
Molecular analysis	Characterizes oil in terms of four constituents using clay-silica gel columns	D2007
Pour point	Temperature at which oil can no longer be poured	D99
Refractive index	Ratio of the velocity of specified wavelength of light in air to the velocity in the oil	D1218 D445
Viscosity (resistance to flow), kinematic	Centistokes (mm^2/sec)	
Saybolt universal second (SUS)	Empirical unit: Time in seconds to flow through standard orifice	D2270
Viscosity index	Empirical dimensionless unit: Measures the effect of temperature change by comparison to standard oils	

Table 3.2 Physical Properties of the Three Base Oil Types

Property	Aromatic	Naphthenic	Paraffinic
Aniline point (˚ C)	15--49	60--99	93--130
Refractive index	1.53--1.58	1.49--1.53	1.46--1.49
Approximate density (API)	10	20	30
Specific gravity	Approx. 1.00	0.91--0.93	0.86--0.88
VGC	0.90--0.999	0.82--0.899	0.79--0.819
Approx. flash point (˚ C)	160	180	230
Molecular weight	246	320	440

oil is compared to standards to which a number has been assigned. Light oil colors have low numbers, and dark oil colors are assigned high numbers.

The color of an oil is generally not a functional property, but there are exceptions. Color tests are sometimes used to determine if automotive transmission fluids have oxidized and require changing. Light oil colors should not be associated with oil purity.

3.2.3 Flash Point (ASTM D92 and D93)

The flash point is the temperature at which a flame will first ignite the vapors above an oil. There are a number of procedures that give different results, but perhaps the most commonly used in the lube oil business is the Cleveland Open Cup (COC). The flash point correlates closely with the beginning of an oil's boiling range. Flash point depression is very noticeable when oils of a wide range of viscosities are blended together to produce an oil of intermediate viscosity, a so-called dumbbell blend. For example, if an oil with a viscosity of 150 cSt at 40° C (ISO 150) were desired, a number of different blending schemes could be used. If oils of 100 cSt and 200 cSt at 40° C were used, a much higher flash point would be achieved than if oils of 20 cSt and 500 cSt were used. In the blends, the flash point will be very close to the flash point of the low-viscosity component. At any given viscosity, paraffinic oils have a higher flash point than naphthenics. This is a reflection of the higher molecular weight of paraffinic oils at a given viscosity. Flash point is important for safety reasons, as too low a flash point can cause fire problems.

3.2.4 Gravity or Density (ASTM D287, D1298, and D2501)

A basic characteristic of an oil is its density. Density is defined as the mass of a unit volume of material--;grams per cubic centimeter or pounds per cubic foot. Another common measure of this property is the specific gravity, which is the ratio of the density of a material to the density of water. Both densities are measured at the same temperature and expressed in the same units. Specific gravity is a dimensionless (unitless) number.

Another measure that is widely used in the oil industry is the API gravity (Equation 3.1). Historically this measure has been commonly used in the production and refining of oils. It is inversely related to specific gravity, from which it can be calculated.

$$\text{API gravity (}^\circ\text{API)} = \frac{141.5}{\text{specific gravity}} - 131.5 \tag{3.1}$$

As the specific gravity increases, the API gravity decreases. Since density and gravity are very temperature-dependent, it is necessary to specify the temperature at which they are measured. The standard reference temperature in the oil industry is 60°F (15.6°C). Lube base oils commonly range from about 15 to 35 API depending upon type and viscosity.

Oil density increases as the number of ring structures in the oil increase, because ring structures can be packed closer together than straight-chain hydrocarbons. Naphthenic oils have more ring structures than paraffinic oils which results in them having a higher density or a lower API gravity than paraffinic oils of the same viscosity. As viscosity increases, density increases and API gravity decreases within a related product line. A typical example of the gravity change with oil type is shown in Table 3.2.

API gravity is an important commercial property because oils are sold by volume at 60°F. THe API gravity and temperature are measured simultaneously, and appropriate correction factors are applied to determine the volume at 60°F.

Since density and API gravity for an oil will vary with viscosity and composition, one way to remove the variable of viscosity is to use the viscosity gravity constant (VGC). The VGC is approximately constant over a wide range of viscosities if the chemical composition is relatively constant. Thus VGC can be used as an indicator of composition. There are some limitations to its use, however, particularly for residual oils and high-asphaltic materials. The formula for calculating VGC is given in ASTM D2501.

3.2.5 Molecular Analysis (ASTM D2007)

The standard molecular type of analysis, described in ASTM D2007, characterizes oil in terms of four constituents by a series of elutions. It is also called a clay-gel test because of the clay and silica gel columns employed. The four constituents determined by this test are asphaltenes, polars, aromatics, and saturates. Asphaltenes are reactive, resinous materials with poor stability. Modern base oils generally do not contain significant quantities of asphaltenes.

Polars are heterocyclic molecules with either a sulfur, nitrogen, or oxygen molecule replacing one of the carbon atoms of the ring system. The ring systems that contain a polar molecule tend to be aromatic, and most are removed in the solvent extraction process. The heterocyclic sites are also very reactive to the addition of hydrogen, so polar content is greatly reduced after hydrogenation. Higher viscosity oils tend to have a higher polar content.

Aromatic molecules have unsaturated, multiple-ring systems. The solvent extraction process will significantly reduce the aromatic content of an oil, but not as much as the process reduces the polar content.

Saturates are that portion of the oil that is not absorbed by the clay and silica gel columns. This portion consists of molecules made up of saturated rings (naphthenes) and varying degrees of side chains (paraffins). Because they are saturated they are relatively inert and thus have stability to heat and light degradation.

This test and other physical tests will give a wealth of information about the composition of an oil. The test will characterize the saturate portion of the oil but cannot determine how much of the saturate consists of naphthenic or paraffinic molecules. Carbon atom analysis is also used to characterize an oil. The percentages of carbon atoms in the aromatic structure (%C_a), naphthenic structure (%C_n), and paraffinic structure (%C_p) can be obtained. Two tests, ASTM D2140 and ASTM D3238, are available that give essentially the same results. These tests are empirical tests that were developed by a correlation with the contents of pure compounds. It is believed that they give a good picture of the "average" oil molecule. The tests are easy to apply, requiring only the measurement of viscosity, density, and refractive index.

3.2.6 Pour Point (ASTM D99)

An oil's pour point can be described as the temperature at which the oil can no longer be poured. The formation of wax crystals and viscosity increase as temperature is lowered.

Paraffinic oils, even though they may have been dewaxed, are still waxy in behavior at low temperatures due to their long straight side chains.

Naphthenic oils made from wax-free crude oil exhibit viscosity increase only at low temperatures, and extremely low pour points are possible.

Pour point depressants are available to retard the formation of wax crystals and thus lower the pour point of an oil. These are polymeric materials that affect only the waxy pour of paraffinic oils. They do not function in naphthenic oils.

3.2.7 Refractive Index (ASTM D1218)

The refractive index (RI) of an oil is the ratio of the velocity of a specified wavelength of light in air to its velocity in the oil. The laboratory procedure for the determination of refractive index is simple and rapid. Refractive index is very sensitive to chemical composition. The speed and sensitivity of the test make it well suited as an incoming quality control tool. The RI will change as viscosity changes, so both parameters are usually measured and recorded. A related parameter, the refractivity intercept, is constant for oils of similar composition. The refractivity intercept is defined as the refractive index minus half the density. The ranges of the refractive index for different compositions of oils appear in Table 3.2.

3.2.8 Viscosity (ASTM D445 and D2270)

Oil viscosity is an important parameter in the lubricating ability of an oil. In simplest terms, viscosity is a measure of resistance to flow. It is related to molecular weight and composition. In the oil industry there are a number of units in which viscosity is commonly measured. Most laboratories now measure the kinematic viscosity in centistokes (cSt) at either 40°C or 100 C. An older and still widely used system of viscosity is the Saybolt universal second (SUS), measured at either 100°F or 210°F.

The centistoke is related to basic units of measurements, and 1 cSt = 1 mm^2/sec. A Saybolt universal second is an empirical unit and is defined as the time in seconds required for a given volume of oil to flow through an orifice of given size at a given temperature. Tables exist for converting one unit of measurement to the other.

Lube oil viscosities range from a low of about 3.7 cSt at 40° C or 40 SUS at 100° F to a high of around 50 cSt at 100° C or 260 SUS at 210° F. It is common practice to measure low-viscosity oils at 40° C (or 100° F in the SUS system) and high-viscosity oils at 100° C (or 210° F for SUS).

Dynamic viscosity (measured in poise), Saybolt furol seconds, and Red-wood viscosity are other measures used to define fluid viscosity. Another common classification system is the ISO viscosity grade number, which is the nearest whole number to the center of the viscosity range for a given oil. The viscosity is measured in centistokes at 40° C (see Table 3.3).

The viscosity of an oil is temperature-dependent, and precise control of temperature is mandatory when measuring viscosity.

The viscosity index (VI) is an empirical unitless number that was developed to define and quantify the natural tendency of oil to change viscosity with temperature. The VI is determined by comparing the viscosity of the oil sample to two standard oils having arbitrary VIs of 0 and 100. The graphical and computational methods to determine VI are explained in ASTM D2270. Low VI numbers indicate that the oil viscosity is sensitive to temperature change.

The VI of an oil is dependent upon its chemical composition. Straight-chain hydrocarbons have less tendency to change viscosity with temperature and will have high VI numbers. Cyclic hydrocarbons, both saturated and unsaturated, show greater change with temperature and will have low VI numbers. Paraffinic oils have an inherently higher VI than naphthenic oils. Additives known as VI improvers are available that will increase the VI of an oil formulation and reduce the amount of viscosity change as temperature changes.

The viscosity index was originally developed for use with motor oils, for which a viscosity that remains relatively constant with varying temperature is desired. For instance, an engine oil must flow freely enough to permit cold starting but must be viscous enough to provide lubrication at operating temperature. VI is also a functional property in some hydraulic oil applications.

Table 3.3 International Organization for Standardization (ISO) Viscosity Classification System for Industrial Fluid Lubricants

	Viscosity grade range (cSt at 40°C)	
ISO viscosity grade number	Minimum	Maximum
2	1.98	2.42
3	2.88	3.52
5	4.14	5.06
7	6.12	7.48
10	9.00	11.0
15	13.5	16.5
22	19.8	24.2
32	28.8	35.2
46	41.4	50.6
68	61.2	74.8
100	90.0	110
150	135	165
220	198	242
320	288	252
460	414	506
680	612	748
1000	900	1100
1500	1350	1650

Paraffinic oils are commonly available with VI values of 100 or greater. Naphthenic oils show a wider variation, with an upper value of 65, but VI is not usually an important functional characteristic for naphthenic oils.

3.2.9 Base Oil Selection

When selecting a base oil for a lubricant, the application conditions must be considered. The lubricant must provide an unbroken film between the sliding metal surfaces to minimize wear without excessive frictional forces and heat. It must absorb and transfer the heat away from the moving surfaces and be stable and maintain its properties throughout the range of application temperatures and other operating conditions. It must also be economical to use.

Perhaps the most important single characteristic in selecting the proper oil is viscosity. To achieve the most efficient reduction in friction, an oil film must be formed that separates the moving surfaces. To maintain this separation, the oil must be able to resist the pressure applied to it. Thus, higher load-bearing requirements call for a higher viscosity oil. However, higher viscosity oils require more force to move or shear them. Therefore, to achieve the lowest coefficient of friction, you need oil of the lowest viscosity that will maintain a full film under the conditions encountered.

Two other important parameters to consider are temperature and speed. The temperature the oil encounters in operation will determine its viscosity. An oil that thins out too much at high temperature will not maintain a safe film. Higher temperatures of operation require a higher viscosity oil. At the other extreme, low-temperature operations require a low-viscosity oil with a low pour point to prevent excessive thickening. Higher operating temperatures also affect the stability of an oil. Prolonged high-temperature operation can degrade the oil, causing an increase in viscosity and a buildup of carbon, sludge, and other breakdown products. Additives are available to improve temperature stability. Highly saturated oil molecules are more heat-stable, and in extreme cases synthetic oils may be required.

The speed of the moving parts being lubricated will affect the viscosity requirements. Higher surface speeds require low-viscosity oils with sufficient adhesion to cling to the rotating surfaces despite increased centrifugal force. High-speed operations also require high oil flow, which can be achieved with lower viscosity oils.

Aromatic oils are not considered candidates for lubricating oils. The advantages of naphthenic and paraffinic oils are listed in Table 3.4.

3.3 ADDITIVES

Base oils do not always have the properties required by the application. Manufactured chemicals known as additives are added to the base oils to

Table 3.4 Advantages of Naphthenic and Paraffinic Lubricating Oils

Naphthenic base oil	Paraffinic base oil
Solubility: Additives will dissolve more readily into a naphthenic base oil.	*High VI*: Paraffinic oils give smaller viscosity changes with temperature than naphthenic oils.
Emulsification: Much easier to emulsify and form a soluble oil than a paraffinic oil.	*Demulsibility*: Much easier to separate water from paraffinic oil than from naphthenic oils.
Low-temperature performance: Better than waxy paraffinic and is essential for winter use and refrigeration.	*High-temperature properties:* More heat-stable than naphthenic oil.

Table 3.5 Quantity of Additive for Various Applications

Application	Weight %
Gear oil	5--50
Marine cylinder oil	10--30
Zinc-free railroad oil	7--20
Automotive engine oil	10--20
Diesel engine oil	7--15
Hydraulic fluid	4--7
Industrial lubricants	0.5--2

provide the required performance. The amounts of additives required will depend upon the application (Table 3.5). It is estimated that over 7% of the 10 billion gallons of finished lubricants used annually are additives that enhance properties. Types of additives, their purpose, and their typical compositions are summarized in Table 3.6.

3.3.1 Antifoamants

The motion within gearboxes, engines, and other machinery churns and aerates the lubricating oil. The air can become trapped in the oil and create foam, which reduces the lubricating properties and may result in severe damage to the machinery. Silicone polymers are often added to the oil to reduce the surface tension and allow the foam bubbles to collapse.

3.3.2 Corrosion Inhibitors

As oils age in the application, they become oxidized, and acids are formed that may attack and corrode metal surfaces. Overbased (high pH value) metallic sulfonates, phenates, and fatty amines are added to the oil to neutralize and prevent the formation of the acids that attack metallic surfaces.

3.3.3 Dispersants and Detergents

As oils oxidize and age, they can leave hard carbon deposits or sludge on engine and gear components. These deposits will break down lubricating films and cause catastrophic mechanical failure. The sludge deposits can also form on oil seal lips and destroy the sealing mechanism, which will result in leakage. A number of chemicals (succinimides, neutral metallic sulfonates, phenates, polymeric detergents, amine compounds) can be added to the oil to keep the surfaces clean and keep the carbon particles in suspension. These particles are thrown away when the oil is changed.

Table 3.6 Additives for Lubricating Oils

Type	Purpose	Typical chemical compounds
Antifoamants	Reduces foam in crankcase	Silicone polymers
Corrosion inhibitors	Neutralize acids, prevent corrosion from acid attack	Overbased (high pH) metallic sulfonates, phenates, fatty amines
Dispersants, detergents	Keep sludge, carbon, and other deposits suspended in the oil	Succinimides, neutral metallic sulfonates, phenates, polymeric detergents, amine compounds
Extreme pressure (EP) antiwear additives	Form protective film on engine parts, reduce wear, prevent scuffing and seizing	Zinc dialkylidithiophosphates; tricresyl phosphates, organic phosphates, chlorine and sulfur compounds
Friction modifiers	Reduce coefficient of static friction between metal surfaces	Long-chain polar compounds (amides, phosphates, phosphites, acids)
Metal deactivators	Form surface films so that metal surfaces do not catalyze oil oxidation	Zinc dialkyldithiophosphates, metal phenates, organic nitrogen compounds
Oxidation inhibitors	Prevent or control oxidation of oil, formation of varnish, sludge, and corrosive compounds; limit viscosity increase	Zinc dialkyldithiophosphates, aromatic amines, sulfurized products, hindered phenols
Pour point depressants	Lower "freezing" point of oils, ensuring free flow at low temperatures	Low molecular weight methacrylate polymers
Rust inhibitors	Prevent the formation of rust on metal surfaces by formation of surface film or neutralization of acids	High base additives, sulfonates, phosphates, organic acids or esters, amines
Seal compatibility enhancer	Maintain softness and pliability of seal material	Phosphate esters, phenols, aromatic esters, sulfones or lactones
Viscosity index improvers	Reduce the rate of viscosity change with temperature; reduce fuel consumption; maintain low oil consumption; allow easy cold starting	Polyisobutylene, methacrylate, acrylate polymers, olefin copolymers; may incorporate dispersant groups

3.3.4 Extreme Pressure (EP) Antiwear Additives

When gears mesh at high speeds, extremely high local pressures are generated at the gear teeth, and lubricants break down. Extreme pressure (EP) additives are put in gear oils to provide a protective film on the gear teeth. The compounds (zinc dialkyldithiophosphate, tricresyl phosphate, organic phosphate, chlorine and sulfur compounds) plate out and form protective metal salts on the gear teeth when the pressure and temperature increase. These salts prevent gear tooth wear, but they can also plate out between the seal lip and the shaft. This may result in heavy shaft and lip wear and early leakage. Reducing the force between the seal lip and the shaft will help prevent the plating from occurring.

3.3.5 Friction Modifiers

Oily metal surfaces tend to stick together when machinery is at rest. Excessive start-up torque may result unless friction modifiers are added to the base oil. The friction modifiers are long-chain polar compounds (amides, phosphates, phosphites, acids, etc.) that plate out on the metal surfaces and reduce the coefficient of static friction. This reduces torque and frictional heating at the oil/metal interface and provides better lubrication under boundary conditions.

3.3.6 Metal Deactivators

Hot metal surfaces can heat the oil and cause rapid oxidation and sludging. This process is accelerated when the metal surface acts as a catalyst. Zinc dialkyldithiophosphates, metal phenates, and organic nitrogen compounds are added to oils to deactivate the metal surfaces and prevent oil oxidation.

3.3.7 Oxidation Inhibitors

Oxygen reacts with oil molecules at high temperatures and can polymerize the oil, which creates sludge, increases oil viscosity, and reduces lubrication. Thick oil will reduce fuel economy and create starting difficulties in cold weather. Aromatic amines, zinc dialkyldithiophosphate, sulfurized products, and hindered phenols prevent the oxidation of base oils.

3.3.8 Pour Point Depressants

Cold weather will cause large lacy wax crystals to form in some oils, which increases viscosity and results in high starting torque. Low molecular weight methacrylate polymers are dissolved in light oils and added to the base oil to lower the freezing or pouring point of the oil. A small amount (0.25--0.5%) of the undiluted polymer can lower the freezing point of most base oils by 10--20°C (18--36°F).

3.3.9 Rust Inhibitors

Rust on metal surfaces is prevented by adding sulfonates, phosphates, organic acids, esters, or amines to the base oil. These chemicals coat the metal surfaces with a protective film or neutralize acids.

3.3.10 Seal Compatibility Enhancer

Some rubber compounds made with nitrile (NBR) or polyacrylate (PA) polymers will harden when exposed to oils and oil additives at elevated temperatures. When the seal lip becomes hard, it loses its compliance and cannot follow shaft vibrations, and leakage often results. Chemicals (phosphate esters, phenols, aromatic esters, sulfones, lactones) are often added to the oil to soften the rubber material and regain its compliance. These additives are sometimes added to the oil after leakage has begun. In some cases, these chemicals may react adversely with the seal material and cause catastrophic leakage. It is recommended that seal materials be upgraded to prevent hardening. It is not recommended to add chemicals to an oil to soften hardened seal lip materials.

3.3.11 Viscosity Index Improver

Many oils are required to perform consistently over a broad temperature range. Straight mineral oils cannot meet these requirements because their viscosity changes rapidly with temperature. Acrylate polymers, olefin co-polymers, and other oil-soluble polymers are added to a low-viscosity base oil to increase the viscosity at high temperatures (Fig. 3.3). This prevents the lubricating oil film from becoming too thin to protect the moving metal surfaces.

Manufacturers of lubricating oils may use different quantities of various additive chemicals to make an oil meet specifications. Seal manufacturers and users cannot assume that a seal material that functions well in an oil from one manufacturer will function equally well in an oil from another supplier even though the oils are designed to meet the same test. If any doubts exist, then the seal material should be immersion-tested in each oil. In some cases, seal samples should be functionally tested in each oil at typical application conditions.

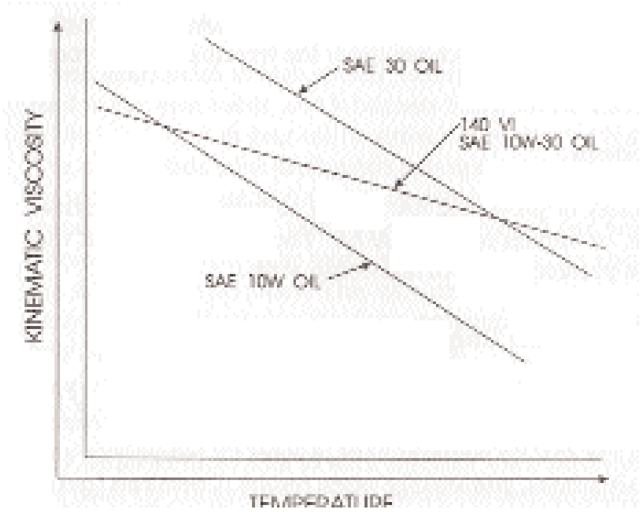


Fig. 3.3 Viscosity index improvers increase base oil viscosity at high temperatures.

3.4 LUBRICATING GREASES

Lubricating grease is made by adding a thickening agent (metallic soap, urea compound, carbon black, organophilic modified clay, or other material) to a mineral-based oil or a synthetic fluid. Additives are also often added to improve performance. Greases are used in applications where the lubricant must stay in its original position in the mechanism. Grease helps prevent external contaminants from entering the gear box and reduces noise from gear boxes.

3.4.1 Grease Properties

The key properties that define a grease are listed in Table 3.7. Apparent viscosity is a term used to describe the resistance to shear of a non-Newtonian fluid such as a grease (ASTM D1092). The viscosity varies with shear rate, and at very low shear rates the grease acts as a solid or semi-solid. When shear rates increase, the viscosity approaches that of the base fluid. The apparent viscosity of a grease is affected by the type and amount of thickener used and cannot be lower than the base fluid viscosity. This property is used to determine how well the grease will flow through dispensing lines and predict its performance in the application. Consistency is determined by measuring the depth in tenths of a millimeter to which a standard cone will penetrate a sample of the grease under specified condi-

Table 3.7 Measurement of Grease Properties

Property	ASTM test	Description
Apparent viscosity	D1092	Measures ratio of shear stress to shear at a given temperature and shear rate. Varies with shear rate as grease is a non-Newtonian fluid.
Consistency	D217, D1403	Measures relative hardness using cone to penetrate the grease.
Compatibility with elastomers	D4289	Immersion test to determine effect of grease on seal materials.
Corrosion resistance	D1743	Determines the damage done to metallic surfaces exposed to the test grease at 100% relative humidity at a given temperature.
Dropping point	D566, D2265	Determines temperature when grease passes from plastic to liquid state.
Evaporation	D972, D2595	Determines weight loss due to evaporation at a given temperature.
Flow	D3232	Determines flow properties at high temperatures and low shear rates.
Functional	D1263	Measures leakage tendencies from unsealed wheel assembly.
	D1264	Measures resistance to water washout in rotating bearings.
	D1741	Determines life of grease-lubricated 306 ball bearings at 3600 rpm at a given service temperature.
	D3527	Determines high-temperature life of wheel bearing grease.
	D4290	Measures leakage tendencies from unsealed wheel bearing assembly at high temperatures (160˚ C or more).
Load-carrying ability	D2509	Determines load-carrying ability of lubricating grease with Timken test.
	D2596	Determines load-carrying ability of lubricating grease with four-ball tester.
Low-temperature torque	D1478	Determines how grease retards ball bearing at low temperatures.
	D4693	Determines how grease retards wheel bearing assembly at low temperatures.
Mechanical stability	D1831	Determines consistency change after working in . tester
Oxidation stability	D942	Static, sealed test at elevated temperature measures grease stability when exposed to oxygen.
Structural stability	D1742	Determines tendency of oil to separate from grease under static conditions.
Wear resistance	D2266	Four-ball test to measure grease wear-preventive characteristics.
	D4170	Measures fretting wear protection of grease.

Table 3.8 NLGI Consistency Number for Grease Using ASTM D217 Test Procedure

NLGI number	ASTM worked penetration (60 strokes) at 25° C (77° F) in tenths of a millimeter
000	445--475
00	400--430
0	355--385
1	310--340
2	265--295
3	220--250
4	175--205
5	130--160
6	85--115

Source: National Lubricating Grease Institute.

tions (ASTM D217, ASTM D1403). The National Lubricating Grease Institute (NLGI) classifies grease in grades that depend upon the depth of penetration (Table 3.8).

Compatibility with seal elastomers is very important to ensure that the materials maintain their sealing ability. Excessive grease leakage or ingress of contaminants can lead to gross failure. The hardness and volume changes of elastomers are typically measured after immersion in the test grease at the specified temperature (ASTM D4289). Greases can sometimes corrode metallic surfaces when stored in high-humidity areas after the initial run. This corrosion can damage bearings and result in shortening their life. Greases should be tested to ensure that they are compatible with the metal surfaces used in the application (ASTM D1743). As greases are heated, they become liquefied. ASTM D566 describes a test to determine the temperature [up to 260° C (500° F)] at which the grease becomes liquid. This temperature is known as the drop point. The ASTM D2265 test procedure will determine the drop point for high-temperature grease [up to 330° C (625° F)]. ASTM D972 and ASTM D2595 describe test procedures to determine the weight loss of a grease due to evaporation at temperatures upto 316° C (600° F).

The flow properties of lubricating greases under high-temperature, low-shear conditions can be used to determine the lubricating conditions in a bearing as well as to predict how well the grease can be transmitted in dispensing systems (ASTM D3232).

Various functional tests nave been developed to determine how grease properties affect bearing performance. ASTM D1263 is a 6 hr screening test that evaluates the tendency of a grease to leak from an unsealed wheel bearing assembly at 104˚ C (270˚ F). A higher temperature test for unsealed wheel bearings is described by ASTM D4290. The test is run for 70 hr at 1000 rpm at 160˚ C (320˚ F) with a thrust load of 111 N (25 lb). Another screening test determines the performance of a standard 306 ball bearing with the test grease at temperatures up to 125˚ C (257˚ F) at 3600 rpm (ASTM D1741). High-temperature life tests for wheel bearing grease are described by ASTM D3527. The ability of a lubricating grease to resist water washout during operation can be determined with ASTM D1264.

Bearings and grease are often rated by their load-carrying capabilities. ASTM D2509 describes a method to determine the load-carrying capabilities of a grease using a special test machine designed by the Timken Corporation. A four-ball tester can also be used to determine the load-carrying ability (ASTM D2596) and the wear resistance (ASTM D2266) of the grease. The extent to which the grease retards the rotation of a slow-speed ball bearing at temperatures below 18˚ C (0˚ F) can be determined with ASTM D1478. Low-temperature testing has also been extended to wheel bearing assemblies (ASTM D4693).

The mechanical stability of a grease is determined by measuring changes in its consistency after working or shearing the grease in a tester for 2 hr at room temperature (ASTM D1831). The stability of a grease to oxygen is determined by placing the grease in a sealed static container filled with oxygen at an elevated temperature (ASTM D942). This test is used to compare the oxidation resistance of various greases and cannot be used to predict performance. ASTM D1742 describes a procedure to determine the tendency of the base oil to separate from the grease while stored in a container. The test is suitable for NLGI #1 of harder greases and cannot be used to predict separation during dynamic conditions.

When ball bearings are installed in machinery and then shipped for long distances, the vibration can cause fretting wear. ASTM D4170 evaluates how well a grease can protect a bearing from fretting wear when the bearing is oscillated under load.

3.4.2 NLGI (ASTM Designation, Description, and Performance Requirements)

The Society for Automotive Engineers (SAE), the National Lubricating Grease Institute (NLGI), and the American Society for Testing and Materials (ASTM) have developed a system to classify and describe the required

Table 3.9 NLGI and ASTM Designations and Required Performance Tests

NLGI service designation	Intended service	NLGI service description	ASTM D4950 required performance tests
LA	Chassis	Service typical of chassis components and universal joints in passenger cars, trucks, and other vehicles under mild duty only.a	P, DP, 4BW, EC
LB	Chassis	Service typical of chassis components and universal joints in passenger cars, trucks, and other vehicles under mild to severe duty[b]	P, DP, OS, RP, 4BW, 4BEP, FW, EC, LTL
GA	Wheel bearings	Service typical of wheel bearings operating in passenger cars, trucks, and other vehicles under mild duty	P, DP, LTL
GB	Wheel	Service typical of wheel bearings bearings operating in passenger cars, trucks, and other vehicles under mild to moderate duty[c]	P, DP, WW, OS, RP, 4BW, HTL, EC, L, LTL
GC	Wheel bearings	Service typical of wheel bearings operating in passenger cars, trucks, and other vehicles under mild to severe duty	P, DP, WW, OS, RP, 4BW, 4BEP, HTL, EC, L, LTL

P = Penetration D217; DP = dropping point D566 or D2265; WW = water washout D1264; OS = oil separation D1742; RP = rust protection D1743; 4BW = four-ball wear D2266; 4BEP = four-ball EP D2596; HTL = high-temperature life D3527; FW = fretting wear D4170; EC = elastomer compatibility D4289; L = leakage D4290; LTL = low-temperature torque D4693.

[a] Mild duty will be encountered in vehicles operated with frequent relubrication in noncritical applications.

[b] Severe duty will be encountered in vehicles operated under conditions that may include prolonged relubrication intervals, or high loads, severe vibration, exposure to water or other contaminants, etc.

[c] Moderate duty will be encountered in most vehicles operated under normal urban, highway, and off-highway service.

Source: National Lubricating Grease Institute.

performance tests for automotive greases (ASTM D4950). The service designation, intended service, service description, and required ASTM performance tests are summarized in Table 3.9. LA and LB are service designations for chassis applications, while GA, GB, and GC indicate wheel bearing applications. A symbol has been developed that identifies what service capabilities the grease is rated for (Fig. 3.4). Dual symbols

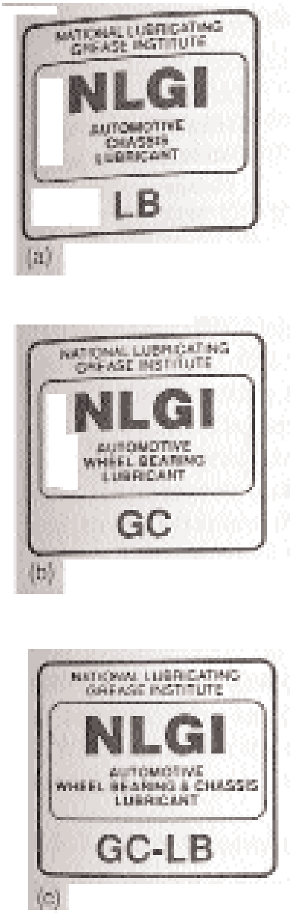

NATIONAL LUBRICATING GREASE INSTITUTE
NLGI
AUTOMOTIVE CHASSIS LUBRICANT
LB
(a)
NATIONAL LUBRICATING GREASE INSTITUTE
NLGI
AUTOMOTIVE WHEEL BEARING LUBRICANT
GC
(b)
NATIONAL LUBRICATING GREASE INSTITUTE
NLGI
AUTOMOTIVE WHEEL BEARING & CHASSIS LUBRICANT
GC-LB
(c)

Fig. 3.4 NLGI symbols for labeling automotive greases, (a) Symbol for automotive chassis application, (b) Symbol for wheel bearing application, (c) Symbol for both chassis and wheel bearing application. (Courtesy of National Lubricating Grease Institute.)

(Fig. 3.4c) indicate that the grease can be used for both chassis and bearing applications.

3.4.3 Application Guide

The relative importance of grease properties for specific automotive applications is shown in Table 3.10 (SAE J310). A grease application guide has

Table 3.10 Relative Importance of Lubricating Grease Properties for Automotive Uses Shown

Property	Wheel bearings	Universal joints	Chasis	Extended lubrication, internal chassis	Multi-purpose applications
Structural stability (including mechanical stability)	H	M	L	H	H
High dropping point (high-temp. service)	H	M	L	M	H
Oxidation resistance	H	M	L	M	H
Protection against friction and wear	M	H	M	H	H
Protection against corrosion	M	M	L	H	M
Protection against washout	M	M	M	H	M

H = Highest; M = moderate; L = least.
Source: Society of Automotive Engineers (SAE J310).

been prepared by NLGI that compares the properties of common greases with different thickeners (Table 3.11).

3.5 ENGINE OILS

3.5.1 API Service Classification

Oils formulated for use in gasoline and diesel fuel must lubricate moving parts; remove soot, varnish, water, and other contaminants from the engine; and help prevent combustion gases from blowing past the gaps between the piston, piston rings, and cylinder walls. The American Petroleum Institute (API) has established a system to rate engine oils by performance levels or application categories.

Oils designed for gasoline engines (Table 3.12) have a designation starting with S (service), and the diesel engine oil designation (Table 3.13) begins with C (commercial). Engine oils have evolved considerably over the years, and additive packages have been developed to prevent corrosion, oil oxidation, engine deposits, sludge buildup, and engine wear. The evolution of the oils has also caused an evolution of sealing materials.

Early low-speed, low-temperature gasoline engines used crankshaft seals made from rope or packing material. These seals were replaced by shaft seals made with nitrile and polyacrylate elastomers. Diesel engines used silicone materials. As engines became smaller with higher operating temperatures and speeds, the upgraded oils attacked the seal material and forced a material change to fluoroelastomer for gasoline engines and PTFE for diesel engines. The seal user and manufacturer must ensure that current seal materials will not be degraded by future oils that will be designed to meet the next set of test requirements for new categories (i.e., SH and GG).

3.5.2 Engine Oil Viscosity

SAE J300 [191] describes a system of classifying engine oil viscosity to help engine builders and consumers select the right oil for their applications (Table 3.14). Oils designed for cold weather operation have the letter W after the number and meet the ASTM D2602 test requirements for low-temperature cranking and the low-temperature pumping test described in ASTM D4684. The tests ensure that the oil will flow properly to the oil pump and not generate excessive starting torque at the specified low temperature. Viscosity grades without a W are known as summer grade oils and have viscosities that lie between specified minimum and maximum values. The viscosity grades listed in Table 3.14 define 11 single-grade engine oils. A vehicle owner in the north would be forced to use a summer oil in the warm months and then switch to a winter grade oil (W) during cold weather. Oils can be blended and viscosity index improvers can be added to a single-grade oil to provide a multigrade oil that can be used for all seasons. The single-grade designations are combined; for example, 10W30 means that the oil meets the requirements of a 10W winter grade oil as well as the requirements for a 30 grade summer oil.

3.5.3 Fuel Economy

After the federal government imposed Corporate Average Fuel Economy (CAFE) requirements on the automobile industry, it became obvious that every means of reducing fuel consumption was going to be needed to comply. One area where some savings can be achieved is by reducing engine friction with friction modifiers added to the engine oil. An oil is classified as “Energy Conserving” if testing shows a 1.5% reduction in fuel consumption. If a reduction of 2.7% or greater is achieved, then the oil is classified as “Energy Conserving II” [198].

The Environmental Protection Agency (EPA) has developed a test procedure to determine if an engine oil reduces fuel consumption [192]. Five automobiles representing a cross section of available engines are run on a chassis dynamometer through a driving cycle that is a modified version of a trip through Los Angeles, California. The car’s fuel consumption is measured precisely first using a reference SAE 20W30 reference oil. The fuel economy performance of the candidate oil is compared to the reference oil and is reported as a percentage improvement versus the reference oil.

Table 3.11 Grease Application Guide

Property	Aluminum	Sodium	Calcium, conventional	Calcium, anhydrous	Lithium	Aluminum complex	Calcium complex	Lithium complex	Polyurea	Organo clay
Dropping point (˚ F)	230	325--350	205--220	275--290	350--400	500+	500+	500+	470	500+
Dropping point (˚ C)	110	163--177	95--104	135--143	177--204	260+	260+	260+	243	260+
Max. usable temp. (˚ F)	175	250	200	230	275	350	350	350	350	350
Max. usable temp. (˚ C)	79	121	93	110	135	177	177	177	177	177
Water resistance	Good to excellent	Poor to fair	Good to excellent	Excellent	Good	Good to excellent	Fair to excellent	Good to excellent	Good to excellent	Fair to excellent
Work stability	Poor	Fair	Fair to good	Good to excellent	Good to excellent	Good to excellent	Fair to good	Good to excellent	Poor to good	Fair to good
Oxidation stability	Excellent	Poor to good	Poor to excellent	Fair to excellent	Fair to excellent	Poor to good	Fair to excellent	Good to excellent	Good	
Protection against rust	Good to excellent	Good to excellent	Poor to excellent	Poor to excellent	Poor to excellent	Good to excellent	Fair to excellent	Fair to excellent	Fair to excellent	Poor to excellent

Pumpability in centralized systems	Poor	Poor to fair	Good to excellent	Fair to excellent	Fair to excellent	Fair to good	Poor to fair excellent	Good to excellent	Good to	Good
Oil separation	Good	Fair to good	Poor to good	Good	Good to excellent	Good to excellent	Good to excellent	Good to excellent	Good to excellent	Good to excellent
Appearance	Smooth and clear	Smooth to fibrous	Smooth and buttery	Smooth and buttery	Smooth and buttery	Smooth and buttery	Smooth and buttery	Smooth and buttery	Smooth and buttery	Smooth and buttery
Other properties		Adhesive and cohesive	EP grades available	EP grades available	EP grades available, reversible	EP grades available, reversible	EP and antiwear inherent	EP grades available	EP grades available	
Production volume and trend[a]	No change	Declining	Declining	No change	The leader	Increasing	Declining	Increasing	No change	Declining
Principal uses[b]	Thread lubricants	Rolling contact bearings	General uses for economy	Military multiservice	Multiservice automotive and industrial	Multiservice industrial	Multiservice automotive and industrial	Multiservice automotive and industrial	Multiservice automotive and industrial	High-temp (frequent relube)

[a] Lithium grease over 50% of production and all other below 10%.

[b] Multiservice includes rolling contact bearings, plain bearings, and others.

Source: Reprinted with permission from the NLGI Lubricating Grease Guide, Copyright 1984, National Lubricating Grease Institute.

Table 3.12 API Gasoline Engine Oil Service Classification Systems

Letter design-ation	API engine service description	Status ASTM requirements
SA	Denotes service typical of older engines operated under such mild conditions that the protection afforded by compounded oils is not required. This category has no performance reequirements, and oils in this category should not be used in any engine unless specifically recommended by the equipment manufacturer. . .	Obsolete Oil without additive No test required.
SB	Denotes service typical of older engines operated under such mild conditions that only minimum protection afforded by compounding is desired. Oils designed for this service have been used since the 1930s and provide mild antiscuff capability and resistance to oil oxidation and bearing corrosion. They should not be used in any engine unless specifically recommended by the equipment manufacturer.	Obsolete test techniques. Provides some antioxidant and antiscuff capabilities.
SC	Denotes service typical of gasoline engines in 1964--1967 models of passenger cars and some trucks operating under engine manufacturers' warranties in effect during those model years. Oils designed for this service provide control of high-and low-temperature deposits, wear, rust, and corrosion in gasoline engines.	Obsolete test techniques (1964--1967).
SD	Denotes service typical of gasoline engines in 1968--1970 models of passenger cars and some trucks operating under engine manufacturers' warranties in effect during those model years. This category may also apply to certain 1971 and/or later models as specified (or recommended) in the owner's manuals. Oils designed for this service provide more protection against high- and low-temperature engine deposits, wear, rust, and corrosion in gasoline engines than oils that are satisfactory for API Engine Service Category SC and may be used when API Engine Service Category SC is recommended.	Obsolete test techniques (1968--1971).

SE	Denotes service typical of gasoline engines in passenger cars and some trucks beginning with 1972 and certain 1971--1979 models operating under engine manufacturers' warranties. Oils designed for this service provide more protection against oil oxidation, high-temperature engine deposits, rust, and corrosion in gasoline engines than oils that are satisfactory for API Engine Service Categories SD or SC and may be used when either of these categories is recommended.	Obsolete test techniques (1972--1980).
SF	Denotes service typical of gasoline engines in passenger cars and some trucks beginning with the 1980--1988 model years operating under engine manufacturers' recommended maintenance procedures. Oils developed for this service provide increased oxidation stability and improved antiwear performance relative to oils that meet the minimum requirements for API Service Category SE. These oils also provide protection against engine deposits, rust, and corrosion. Oils meeting API Service Category SF may be used when API Service Categories SE, SD, or SC are recommended.	Obsolete test requirements (1980).
SG	Denotes service typical of present gasoline engines in passenger cars, vans, and light duty trucks operating under manufacturers' recommended maintenance procedures. Category SG quality oils include the performance properties of API Service Category CC (certain manufacturers of gasoline engines require oils also meeting the higher diesel engine category CD). Oils developed for this service provide improved control of engine deposits, oil oxidation, and engine wear relative to oils developed for previous categories. These oils also provide protection against rust and corrosion. Oils meeting API Service Category SG may be used when API Service Categories SF, SE, SF/CC, or SE/CC are recommended.	Active test techniques (1989).

Source: American Petroleum Institute (API).

Table 3.13 API Diesel Engine Oil Service Classification Systems

Letter designation	API engine service description	Status
CA	Service typical of diesel engines operated in mild to moderate duty with high quality fuels and occasionally has included gasoline engines in mild service. Oils designed for this service provide protection from bearing corrosion and from ring belt deposits in some naturally aspirated diesel engines using fuels of such quality that they impose no unusual requirements for wear and deposit protection. They were widely used in the late 1940s and 1950s but should not be used in any engine unless specifically recommended by the equipment manufacturer.	Obsolete test techniques. Meets requirements of MIL-L-2104A (1954).
CB	Service typical of diesel engines operated in mild to moderate duty, but with lower quality fuels necessitate more protection from wear and deposits. Occasionally has included gasoline engines in mild service. Oils designed for this service were introduced in 1949. Such oils provide necessary protection from bearing corrosion and from high-temperature deposits in normally aspirated diesel engines with higher sulfur fuels.	Same as above but used in engines with high sulfur fuel.
CC	Denotes service typical of certain naturally aspirated, turbocharged, or supercharged diesel engines operated in moderate to severe duty service and certain heavy duty gasoline engines. Oils designed for this service provide protection from high-temperature deposits and bearing corrosion in these diesel engines, and also from rust, corrosion, and low-temperature deposits in gasoline engines. These oils were introduced in 1961.	Obsolete test technique. Meets requirements of MIL-L-2104B (1964).
CD	Denotes service typical of certain naturally aspirated, turbocharged, or supercharged diesel engines where highly effective control of wear and deposits is vital or when using fuels of a wide quality range, including high-sulfur fuels. Oils designated for this service were introduced in 1955 and provide protection from bearing corrosion and from high-temperature deposits in these diesel engines.	Active test technique. Meets Caterpillar certification requirements (1955).

CD-II	Service typical of two-stroke-cycle diesel engines requiring highly effective control over wear and deposits. Oils designed for this service also meet all performance requirements of API Service Category CD.	Active test techniques. Oil meeting the performance requirement described in category CD by the Cat 1G2 and CRC L--38 tests, plus Detroit Diesel 6V--53 T test to address deposits and wear in turbocharged/supercharged two-stroke-cycle diesel engines.
CE	Service typical of certain turbocharged or supercharged heavy duty diesel engines manufactured since 1983 and operated under both low-speed, high-load and high-speed, high-load conditions. Oils designed for this service may also be used when API Engine Service Category CD is recommended for diesel engines.	Active test techniques. Oil meeting all requirements of category CD and having additional performance as described by the Cummins NTC 400 engine test for oil consumption control and the Mack EO/K/2 engine oil specification.
CF--4	Describes oils for use in high-speed, four-stroke-cycle diesel engines. API CF--4 oils exceed the requirements of the CE category providing improved control of oil consumption and piston deposits. These oils should be used in place of CE oils. They are particularly suited for on-highway, heavy duty truck applications. When combined with the appropriate S category, for example, SG, they can also be used in gasoline- and diesel-powered personal vehicles such as automobiles, light trucks, and vans when recommended by the vehicle or engine manufacturer. Based on L-38, Cat IK, Mack T-6 and T-7, Cummins NTC 400 (revised).	Active test techniques (1990). API CF-4 oils exceed the requirements of the CE category providing improved control of all consumption and piston deposits. These oils should be used in place of CE oils. They are particularly suited for on-highway, heavy duty truck applications. Based on CRC L-38, Cat 1K, Mack T-6 and T-7, Cummins NTC 400 (rev.).

Source: American Petroleum Institute (API).

Table 3.14 SAE Viscosity Grades for Engine Oils (SAE J300)

SAE viscosity grade	Maximum cranking viscosity (cP) at given temperature (˚ C)	Maximum pumping viscosity (cP) at given temperature (˚ C)	Maximum viscosity (cSt) at 100˚ C	Maximum viscosity (cSt) at 100˚ C
0W	3250 at --30	30,000 at --35	3.8	--
5W	3500 at --25	30,000 at --30	3.8	--
10W	3500 at --20	30,000 at --25	4.1	--
15W	3500 at --15	30,000 at --20	5.6	--
20W	4500 at --10	30,000 at --15	5.6	--
25W	6000 at --5	30,000 at --10	9.3	--
20	--	--	5.6	< 9.3
30	--	--	9.3	< 12.5
40	--	--	12.5	< 16.3
50	--	--	16.3	< 21.9
60	--	--	21.9	< 26.1

Source: Society of Automotive Engineers (SAE J300).

3.5.4 Engine Oil Additives

Engine oil additives are often sold as commodities to oil blenders. These additive packages are blended with base oils to make a finished oil that will meet one of the service classifications shown in Tables 3.12 and 3.13, have one of the viscosity grades shown in Table 3.14, and perhaps be classified as “Energy Conserving” or “Energy Conserving II.” The typical engine oil will have a base oil content of 73--85% by weight; VI improvers, 8--15%; and other additives, 7--12% [194]. Typical additive packages for an SF gasoline automotive application and a CD diesel engine application are shown in Table 3.15.

Additives are reactive chemical agents. Many physical and chemical interactions occur among these additives and the base oil that affect the finished oil performance, clarity, viscosity, and other properties. These additives may have an adverse effect on elastomers. It is recommended that seals be tested in new oil formulations.

3.5.5 API Symbol

Most oil containers will display the API symbol that describes the characteristics of the engine oil inside them (Fig. 3.5). This symbol is divided into three areas, each area describing one specific characteristic of the oil. The uppermost part of the symbol displays one of the current API service categories, the center provides the SAE viscosity grade, and the lower

Table 3.15 Typical Additive Packages for Diesel and Gasoline Engine Oils

	Additive (SAE 30) treatment for diesel service		Additive (10W30) treatment for gasoline engine	
Additive purpose	Ingredient	Wt % in oil	Ingredient	Wt % in oil
Dispersant	Polymer amine	6.0	Polymer amine	4.0
Primary detergent	Low base sulfonate	1.5	Low base sulfonate	0.5
Detergent alkaline agent	High base phenate	0.5	--	--
Antioxidant	Low base phenate	1.5	Sulfurized hydrocarbon	0.5
Rust inhibitor alkaline agent	High base sulfonate	0.5	High base sulfonate	1.0
Antioxidant antiwear	Zinc dialkyldithiophosphate	1.5	Zinc dialkyldithiophosphatte A	0.8
Friction reducer	--		Zinc dialkyldithiophosphate B	0.5
Viscosity index improver	--	--	Sulfurized fat	0.7
			Ethylene-propylene copolymer	10.0
Total		11.5		18.0

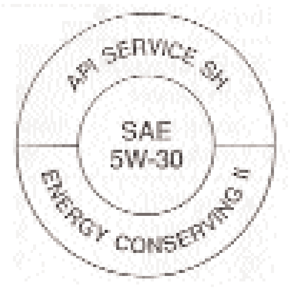


Fig. 3.5 API service symbol for labeling engine oils (Courtesy of American Petroleum Institute.)

section displays the energy conservation designation of the oil. If the lower area is blank, then the oil is not an energy conserver. The display shown in Fig. 3.5 describes an oil for a gasoline engine that will meet the SG service requirements, summer and winter. It has a multigrade viscosity (10W30) that allows use in all climates (down to--20˚ C), and it meets the Energy Conserving II requirement.

3.6 GEAR OILS

3.6.1 API Service Classifications

The API has developed a system of classifying gear oils to help manufacturers and users select the correct oil for a given application (Table 3.16). GL-1 is a mineral oil without additives, and the other classes (GL-2 through GL-6) have additive packages to enhance performance. About 50% of gear oils for North American vehicles are used in heavy truck applications even though these trucks account for only 1% of the total vehicle population.

Heavy truck differentials are much larger than those of passenger cars, and the gear oil is changed more often. Even though passenger cars represent 75% of the North American vehicle population, they consume only 7% of the gear oils. As a result, many gear oils are formulated to meet the heavy truck load requirements and contain EP additives that can be detrimental to seals made with nitrile materials [130b, 235]. Most pinion and some wheel seal applications have switched to fluoroelastomer materials that resist the EP additives.

The most commonly specified and available type of automotive gear lubricant in North America is API GL-5. In Europe and other parts of the world, API GL-4 is used as commonly as API GL-5. The API GL-5 specifi-

Table 3.16 API Service Classification

Classification	Type	Typical application
GL-1	Straight mineral oil	Automotive manual transmission (tractors and trucks), industrial gear boxes
GL-2	Usually contains fatty materials	Worm gear drives, industrial gear boxes
GL-3	Contains mild EP additive	Manual transmissions and spiral bevel final drives
GL-4	Equivalent to obsolete MIL-L-2105 specification usually satisfied by 50% GL-5 additive level	Manual transmissions and spiral bevel and hypoid gears where moderate service prevails
GL-5	Equivalent to present MIL-L-2105D specification; primary field service recommendation of most passenger car and truck builders worldwide	Used for moderate and severe service in hypoid and all other types of gears; also may be used in manual transmissions
GL-6	Same performance as Ford M2C105A/M2C154A	Typically recommended in service conditions where more protection from scoring is desired, e.g., passenger car hypoid axles with high offset

Source: American Petroleum Institute (API).

cation does not address limited slip applications. Factory fill formulations for limited slip differentials are based on performance in the automaker's individual test rig or vehicle. No standard industrywide test is available to evaluate a lubricant's ability to prevent chatter in a limited slip differential.

Industrial gear boxes typically use a gear oil without EP additives (GL-1, GL-2); in some cases they use GL-3, which may contain a mild EP additive. Nitrile seals are usually acceptable in these applications.

3.6.2 SAE Viscosity Grades

Gear oil viscosity grades are defined in SAE J306 [193] and use a system similar to that of engine oils. Grades with a W following the number are considered cold weather oils, and number only grades are summer oils (Table 3.17). Multigrade gear lubricants can be formulated by blending oils and chemicals to provide an all-weather oil. An 80W140 oil has the low-temperature fluidity of an 80W oil as well as the high-temperature characteristics of a 140 grade.

Table 3.17 Axle and Manual Transmission Lubricant Viscosity Classification (SAE J306)

Viscosity grade	Max temperature for viscosity of 150,000 cP (˚ C)	Viscosity at 100˚ C (cSt) Min.	Max.
70W	--55	4.1	--
75W	--40	4.1	--
80W	--26	7.0	--
85W	--12	11.0	--
90	--	13.5	24.0
140	--	24.0	41.0
250	--	41.0	--

Source: Society of Automotive Engineers (SAE J306).

3.7 AUTOMATIC TRANSMISSION FLUIDS

3.7.1 Applications

Automatic transmission fluids (ATFs) are specially formulated to have excellent low- and high-temperature properties. These properties are required to consistently transmit hydrodynamic energy in transmission torque converters under all weather conditions. The fluid must also lubricate bearings and gears, transmit hydrostatic energy in hydraulic systems, transmit sliding friction energy in lubricated bands and clutches, and act as a heat transfer medium to help maintain machinery within the operating temperature range. The fluid may contain as many as 15 different additives to meet all of these requirements.

Automatic transmission fluids are also used in power steering systems, air compressors, and industrial hydraulic systems. Some clutch plate designs require a low start-up friction to prevent sticking at low speeds. Friction-modified transmission fluids such as Dexron II have been developed to meet these requirements. Other clutch plate designs require a non-friction-modified Type F fluid such as Ford M2C33-F to give high frictional drag at low speeds. Silicone seals are often used in automatic transmission applications [141, 142].

3.7.2 Properties of Automatic Transmission Fluids (SAE J311)

Automatic transmission fluids contain corrosion inhibitors to protect metallic surfaces, wear-resistant additives, antifoam agents, antioxidants to mini-

Table 3.18 Typical Values for Automatic Transmission Fluid Properties (SAE J311)

Density		Specific heat [kJ/(kg-° C) at 15° C]	Coeff of thermal expansion [$m^3/(m^3$-° C) . at 15° C]	Maximum low-temperature viscosity		Minimum viscosity (cSt) at 100° C
Temp. (° C)	kg/m^3			Temp. (° C)	cP	
15	850	1.86	0.00072	--18	1,700	5.5
100	820			--23.3	4,000	
150	788			--40	50,000	

Source: Society of Automotive Engineers (SAE J311).

mize sludge, and special additives to ensure smooth transmission operation over a broad temperature range. Typical values for ATFs are listed in Table 3.18.

The fluid cannot perform its required functions if it becomes oxidized. A blotter test is used to determine if excessive oxidation has occurred. A small drop is placed on a blotter. If the drop is small and dark brown, then the fluid should be replaced; if the drop spreads and is red or light brown, the fluid does not require replacing, It should be noted that unused ATF fluids are red.

3.8 SYNTHETIC LUBRICANTS

3.8.1 Synthetic Oils

Mineral oils are organic and decompose rapidly when exposed to elevated temperatures. In many modern applications, temperatures exceed the upper temperature range of mineral oils (225--325° F; 107--163° C) and synthetic lubricants must be used. Most synthetic lubricants remain fluid at low temperatures, and the operating temperature range is extended under cold conditions as well. Synthetic lubes also have more uniform physical properties, less volatility, higher flash points, and lower toxicity than mineral oils. Synthetic lubricants are more expensive than mineral oils and accounted for only 3% of the total lubricating oil market in 1990. This still represents 80 million gallons, and these synthetic lubricants are used in aerospace, industrial, and transportation applications (Table 3.19). Polyalphaolefins (PAOs) represent 37%, organic esters 24%, and all others 39% of the total synthetic lubricant usage.

Polyalphaolefins can be operated continuously from -30 to +157° C (-22 to +347°F). For the various esters (phosphate esters, neopoyol es-

Table 3.19 Synthetic Lubricant Use (80 million gallons in 1990)

Market segment	Typical applications	Percent of total gallons used (1990)
Industrial applications	Gas turbine, hydraulic, gear and bearing oil, metalworking cutting fluids, heat transfer fluid, compressor oil, greases, process oil	50
Transportation	Engine oils, gear oils, greases, transmission fluids	27
Aerospace	Jet turbine lubes, aircraft piston engine oils, hydraulic, greases, coolants	23

ters, and silicate esters), the operating temperature range is --40 to +200˚C (-40 to +392 ˚F). Other key properties for various synthetic lubes are given in Table 3.20.

3.8.2 Synthetic Grease

Some applications operate at high speeds and require precision bearings with high-temperature synthetic grease. Synthetic greases are made by adding thickeners to a synthetic oil similar to the way in which conventional greases are made. The temperature range and bearing speed factor for several synthetic greases are compared to those of a conventional petroleum grease in Table 3.21. The bearing speed factor is defined as the bearing speed in revolutions per minute (rpm) multiplied by the bearing outside bore diameter in millimeters. Even though conventional petroleum-based greases have the highest speed factor (600,000), synthetic grease may be chosen to extend the temperature range.

3.8.3 Seal Selection

The seal material must be compatible with the synthetic lubricant at the elevated application temperatures. Expensive fluoroelastomers and PTFE materials are often used to replace conventional nitriles and polyacrylates. At very low temperatures, these materials may become stiff and unable to follow shaft vibrations. Since synthetic fluids will flow readily at low temperatures, gross leakage may occur until frictional heat warms the seal material. Conventional mineral oil lubricants are very viscous at low temperatures and will not leak even if the elastomer is stiff. Sometimes silicones or fluorosilicones can be used in cold weather applications if they are compatible with the oil.

Table 3.20 Properties of Synthetic Lubricantsa,b

Property	MO	HMO	PAO	NPE	DBE	DB	PB	PAG	SE	PE	FC	S	FS
Viscosity index	F	E	G	G	E	F	F	G	E	F	F	E	E
Low pour point	P	G	G	G	G	G	G	G	E	G	G	G-E	G
Thermal stability	F	G	G	VG	G	G	F	F	G	G	E	G	G
Oxidative stability	F	G	VG	E	G	G	F	G	F	G	R	G	G
Hydrolytic stability	E	E	E	F	F	E	E	G	P	F	E	G	G
Paint compatibility	E	E	E	G	VG	E	VG	M	F	P	G	G	G
Fire resistance	L	L	L	L	L	L	L	L	L	H	H	G	VG
Lubricating ability	G	G	G	G	G	G	G	G	G	G	G	P-G	G
Low volatility	A	G	G	G	A	A	A	G	G	G	VG	G-E	G-E
Radiation resistance	H	H	H	A	A	H	H	A	A	L	H	G	G
Additive solubility	E	G	G	G	VG	E	E	G	F	G	P	P	P
Elastomer swell	L	N	N	M	H	M	L	L	L	H	L	L	L
Compatibility with petroleum lubricant	--	--	G	G	G	E	G	P	F	G	P	P	P
Relative cost	1	2	5	5--10	3--5	5	4	5--10	30--45	5--10	20--30	12--20	20--30

[a] MO = Mineral oil; HMO = super refined petroleum; PAO = polyalphaolefin; NPE = neopolyol ester; DBE = dibasic acid ester; DB = dialkylbenzenes; PB = polybutenes; PAG = polyalkylene glycols; SE = silicate esters; PE = phosphate esters; FC = fluorinated compounds; S = silicones; FS = fluorosilicones.

[b] E = Excellent; VG = very good; G = good; F = fair; P = poor; A = average; H = high; L = low; N = nil.

Table 3.21 Synthetic Grease Properties

Synthetic grease	Temperature range ° C	 ° F	Bearing speed factor (rpm x bearing bore diam. in mm)
Conventional petroleum-based grease	--40 to 121	--40 to 250	600,000
Silicone grease	--73 to 204	--100 to 400	200,000
Ester	--73 to 149	--100 to 300	400,000
Fluoroether	--40 to 288	--40 to 550	400,000
Various synthetic hydrocarbons	--59 to 149	--75 to 300	400,000
Polyether	--40 to 288	--40 to 550	500,000

3.9 COMPATIBILITY TESTING OF ELASTOMERIC MATERIALS IN LUBRICANTS

3.9.1 Nitrite (NBR) in Engine Transmission and Gear Oils

When nitrile compounds are immersed in engine or transmission oils at elevated temperatures, they usually absorb the oil, swell, and soften during the first 50--200 hr. The hardness will reach a minimum as the elastomer absorbs oil and will then increase as the duration of exposure to oil at an elevated temperature lengthens. The hardening rate of different elastomeric materials in various fluids can be compared using the heat resistance index (HRI). HRI is defined as the time required for the hardness to increase by 5 points over the initial value when the elastomer is immersed in a given oil at a given temperature. Materials with a high HRI do not harden as fast as materials with a low HRI and will provide longer life. The hardening characteristics for a given NBR compound will vary considerably from one type of oil to another (Fig. 3.6; Table 3.22). The minimum hardness correlates directly with the amount of oil absorbed by the compound, the amount of softening, and the amount of volume swell (Fig. 3.7). The rate of hardening and volume swell of a given compound in a given oil are important parameters because they can influence seal performance. Excessive material swell can cause the sealing lip inside diameter to become shaft size or greater, which can result in early leakage (in 200 hr or less). Excessive hardening will cause lip cracks, a loss of interference and loss of resiliency. The seal lip will not be able to follow shaft vibrations, and leakage will result.

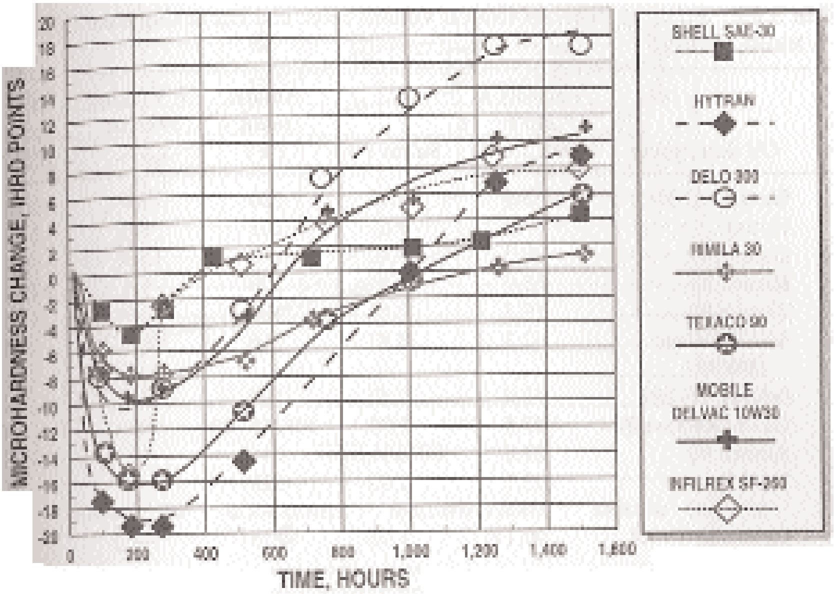


Fig. 3.6 Hardness change in different engine oils at 257° F (125° C) with a given NBR compound. (Courtesy of Chicago Rawhide.)

Most nitriles will harden much sooner in gear oils than in engine oils. Very little softening and volume change will occur, and in some cases no hardness minimum will occur (Fig. 3.8). The additive packages for gear oils are harsher than the additive packages for engine oils; thus compounds that function in engine oils may fail in gear oils.

The effect of changing nitrile compounds in two different gear oils at a constant temperature is illustrated in Fig. 3.9. In gear oil X, nitrile compound A has a 5 point hardness increase after only 25 hr immersion at 225° F (107° C). When the same material is immersed in gear oil Y, the 5 point hardness increase occurs in 260 hr. A material that performs well in an application can thus be severely affected by a slight increase in sump temperature or a change in lubricant. Modifying materials can also improve performance. Nitrile materials A, B, and C have the same initial hardness, but their rates of hardening in gear oils differ. In gear oil X, nitrile material C requires 170 hr to obtain a hardness increase of 5 points, but compound B takes only 40 hr.

The effect of temperature and oil type is illustrated in Fig. 3.10 for a single nitrile compound.

Table 3.22 Effect of Different Oils on Volume Swell and Hardening Rate (HRI) for a Nitrile (NBR)

Oil No.	Oil description	Viscosity at 210˚F (99˚C) (cP)	Sulfur (%)	Aniline point (˚F)	% Volume swell at 125˚	HRI (hr) at 125˚ C
1	Shell SAE 30	63.9	0.40	220	10.6	>1500
2	Hytra	47.5	0.57	205	9.0	1150
3	Delo 300	68.3	0.42	218	15.8	720
4	Rimila 30	65.0	0.42	220	6.	>1500
5	Texaco 90	50.3	0.58	196	7.7	1450
6	Mobil Delvac 10W3	62.3	0.87	216	4.7	850
7	Infilrex SF-26	63.0	0.82	199	5.4	820
8	GL-5 (Rockwell)	87.5	2.22	245	--	124
9	Amoco 90	83.0	1.91	215	--	1000
10	Shell Spirex 85W140	133.7	1.95	223	--	100
11	Renogear H-900	90.1	1.76	215	--	180

--Not measured.
Source: Chicago Rawhide.

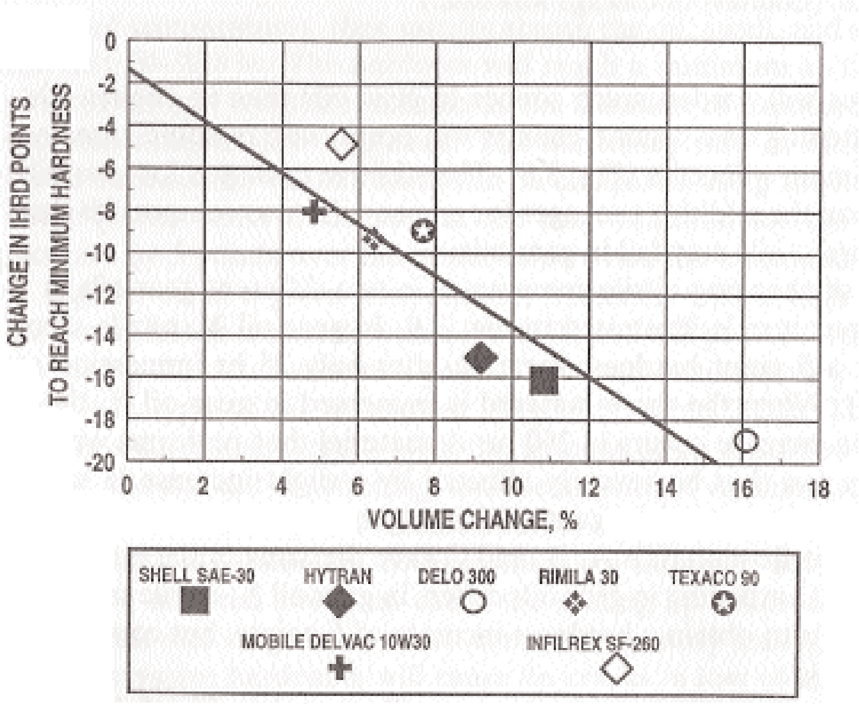


Fig. 3.7 Minimum hardness vs. volume change at 257˚ F (125˚ C) with a given NBR compound. (Courtesy of Chicago Rawhide.)

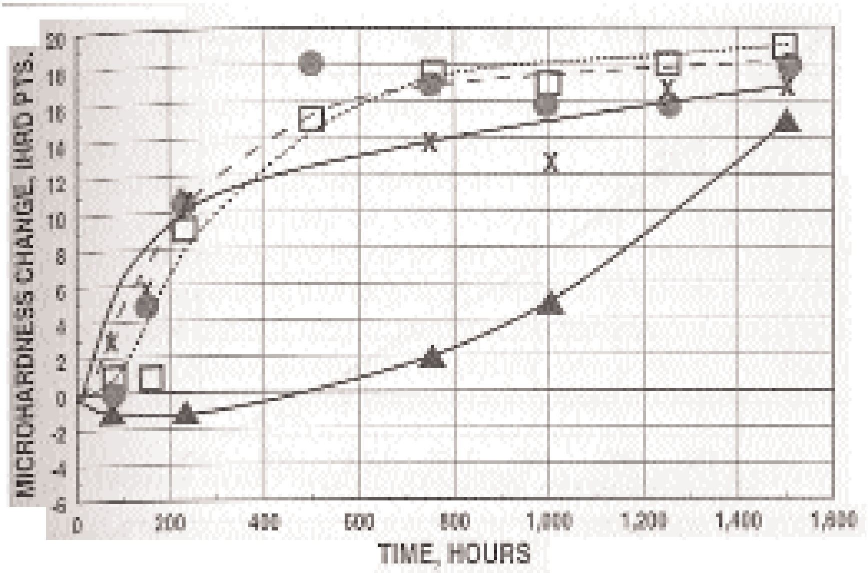


Fig. 3.8 Hardness change in different gear oils at 257˚ F (125˚ C) with a given NBR. (Courtesy of Chicago Rawhide.)

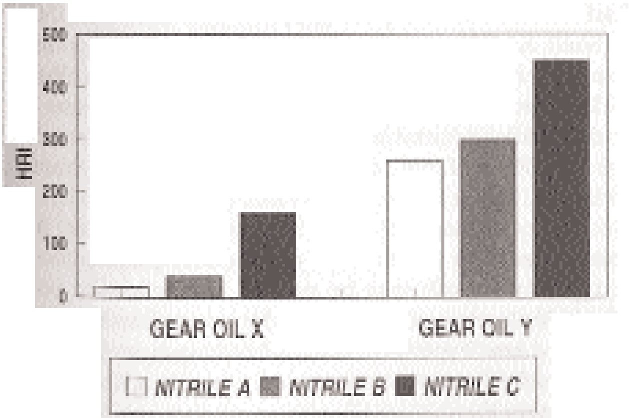


Fig. 3.9 Heat resistance index (HRI) at 225˚ F (107˚ C) for various nitrile compounds in two gear oils. (Courtesy of Chicago Rawhide.)

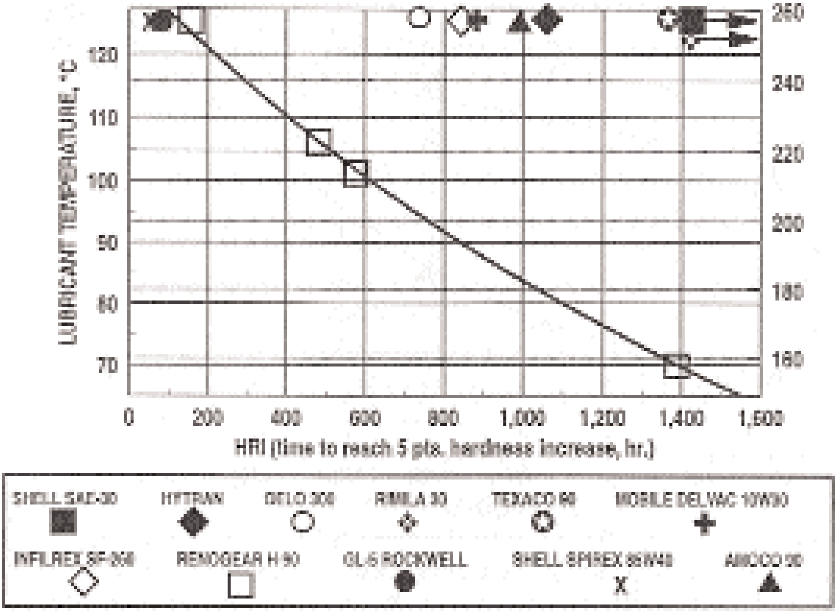


Fig. 3.10 HRI (hr) vs. lubricant temperature for a given NBR compound. (Courtesy of Chicago Rawhide.)

This compound had not achieved a 5 point hardness increase after 1500 hr of immersion in Shell SAE30 and Rimila 30. The same compound reached a 5 point hardness increase after only 100 hr immersion in Shell Spirex 85W140 and 124 hr in GL-5 gear oils. The effect of temperature is clear. At 70˚ C, about 1400 hr is required before a 5 point hardness increase is achieved in Renogear H-90. At 125˚ C, the time to reach the same hardness level is only 180 hr.

Seal suppliers and users must agree on the proper material for the application fluid. In most cases, the original equipment (OE) manufacturer cannot control what replacement oils may be used in an application. An expensive fluid-resistant elastomer is often selected for an application even though a less resistant and less expensive material functions well in the factory fill fluid. The more fluid-resistant compound provides a margin of safety if the end user replaces the factory fill with an oil that can be more aggressive to the original elastomer.

3.9.2 Various Elastomers in Additives

Additives are often added to base oils to improve the lubricant properties and protect machine components. U and P are two different additives that can be added to engine oils to reduce friction. These additives are designed to reduce friction by 40--75%, dissolve sludge, neutralize acid, coat metallic surfaces, reduce wear during cold starts, and increase mileage. They also affect typical shaft seal materials (Tables 3.23 and 3.24). The additives were placed in a base oil, and immersion tests were run at 125° C (257° F) for the nitrile compound and at 150° C (302° F) for all other materials. The results were compared to those of the control oil without additives.

Silicone compounds disintegrated completely within 70 hr. Nitrile, polyacrylate, and ethylene acrylic compounds hardened significantly within a 300 hr period. Fluoroelastomer materials have a mild reaction and should be recommended if friction modifiers are added to engine oils.

3.9.3 Nitrile and Fluoroelastomer in Several Greases

Immersion in grease also affects shaft seal materials. The hardness and volume change of a nitrile and a fluoroelastomer were measured and compared after immersion in greases of several types for 72 hr at 100° C (212° F) (Table 3.25). The property changes of the nitrile material were small when it was exposed to Ronex MP grease but were substantial in the Climax grease. The fluoroelastomer compound did not experience any significant changes in any of the test greases and should be used in place of nitrile when immersion tests show that property changes are excessive.

3.9.4 Resistance of Silicone and Fluoroelastomer 490 to SE and SF Engine Oils

Virtually all engine oils have additive packages to enhance engine life. Two oils with different additive packages (SE, SF) were studied to determine how these packages affect seal materials [117]. SF motor oil caused silicone materials to swell and soften more than SE oil (Table 3.26). The hardness loss (about 25 points) and volume swell (about 22%) for silicone materials after 168 hr at 150° C (302° F) in SF oils was approximately twice that obtained by aging the compounds for 168 hr at the same temperature in an SE motor oil.

The volume change of fluoroelastomer was slight (0--2%) in both SE and SF oils after aging 672 hr at 150° C (302° F). There was virtually no change in hardness. The change in elongation was greater for fluoroelastomer than for silicone. There was no evidence of surface cracking present on any of the samples.

3.9.5 Effect of Humidity on Polyacrylate Seals

Water is not a lubricant, but exposure to humidity will affect some elastorners. Fully cured polyacrylate materials will absorb water when

Table 3.23 Microhardness Change (IRHD) for Various Compounds in Friction Modifiers for Engine Oils

	70 hours				168 hours				336 hours			
Modifier	C	P	U150	U250	C	P	U150	U250	C	P	U150	U250
Nitrile[a]	--6	--9	--6	+8	--5	+18	+20	+20	--4	+19	+26	+26
Polyacrylate	+8	+11	+27	+24	+12	+27	+30	+32	+13	+30	+33	+33
Ethylene acrylic	--4	--2	+8	+8	0	+10	+13	+18	+1	+15	+21	+21
Silicone	--13	D[b]	D[b]	D[b]	--11	D[b]	D[b]	D[b]	--11	D[b]	D[b]	D[b]
Fluoroelastomer	--2	0	0	0	--1	+2	0	0	0	+4	+5	+1

C = Control fluid, Shell SAE 30 without additives; P = additive P, 150 mL per liter of control; U150 = additive U, 150 mL per liter of control; U250 = additive U, 250 mL per liter of control.

[a] Immersion temperature was 125˚ C (257˚ F) for nitrile, 150˚ C (302˚ F) for all other compounds.

[b] D = disintegrated.

Source: Chicago Rawhide.

Table 3.24 Volume Change (%) for Various Compounds in Friction Modifiers for Engine Oils

	70 hours				168 hours				336 hours			
Modifier	C	P	U150	U250	C	P	U150	U250	C	P	U150	U250
Nitrilea	4	6.1	22.3	30.2	4.9	18.8	22.1	29.0	4.6	12.9	21.4	29.0
Polyacrylate	0.3	2.5	--2.7	0.7	0.7	--0.6	--8.9	--6.2	0.8	--2.4	--11.2	--8.7
Ethylene acrylic	8.7	12.2	15.5	12.2	8.6	9.7	6.6	0.6	7.4	2.4	5.3	0.2
Silicone	11.2	D[b]	D[b]	D[b]	11.8	D[b]	D[b]	D[b]	10.8	D[b]	Db	D[b]
Fluoroelastomer	0.4	0.6	9.8	10.6	0.5	8.1	10.2	10.6	0.4	6.7	11.3	12.1

Symbols as in Table 3.23.

[a] Immersion temperature was 125˚ C (257˚ F) for nitrile, 150˚ C (302˚ F) for all other compounds.

[b] Disintegrated.

Source: Chicago Rawhide.

Table 3.25 Property Changes of Nitrile and Ftuoroelastomer after 72 hr in Several Types of Grease at 100˚ C (212˚ F)

	Microhardness change (IRHD)		Volume change (%)	
Grease type	Nitrile	Fluoroelastomer	Nitrile	Fluoroelastomer
Ronex MP Grease	--4	--2	4.7	0.5
MIC45-A --4	--2	9.2	0.4	
Lubriplate 630AA	--10	--1	11.4	0.2
Climax	--12	--4	22.0	1.0

Source: Chicago Rawhide.

Table 3.26 Property Changes of Silicone and Fluoroelastomer Compounds after Immersion in Engine Oil

	Silicone		Fluoroelastomer	
	Shell X100	Shell Fire & Ice 10W40	Shell X100	Shell Fire & Ice 10W40
Property	20W20 (SE)	(SF)	10W20 (SE)	(SF)
Hardness change, IHRD	--9	--25	+4	+2
Tensile change (%)	+12	--4	--31	--3
Elongation change (%)	--5	--4	--46	--26
Volume change (%)	10.7	21.2	0.0	+0.9

Source: Chicago Rawhide.

exposed to high humidity. The material will soften, and the seal hardness (Sec. 2.3.1), radial load, and lip opening pressure (LOP) (Sec. 5.4.22) will decrease. When exposed to a dry atmosphere, the seal material will release the water. The hardness, radial load, and LOP of fully cured shaft seals will increase to nearly the original values. The process is reversible (Table 3.27) and can create problems when hardness, sprung interference (Sec. 4.2.4), radial load, and LOP are used as quality control parameters. A study was conducted to determine the effect of humidity on these parameters [81].

A group of 11 seals from a single production lot for a 2 in. shaft were aged 24 hr at 100% relative humidity. The hardness, sprung radial load, lip interference, and LOP were measured. The measurements were repeated after the same seals had aged in 52% and 0% relative humidity for 96 hr.

The sprung interference is virtually independent of humidity. The radial load varies significantly with humidity, but none of the measurements of

Table 3.27 Reversible Effect of Humidity on Polyacrylate Seal Physical Properties[a]

	Sequence				
Property	100% RH	0% RH	52% RH	0%RH	100% RH
Microhardness (spec 70 ±5)	55	78	69	77	63
Lip opening pressure (psi) (spec 8.5 ±2.5)	6.6	11.2	8.7	10.9	7.1
Radial load (oz/in.) (spec 11 ±3.3)	9.7	15.6	10.9	13.2	9.0

[a] Exposure for 70 hr at each humidity level (350 total hours).

Source: Chicago Rawhide.

the seal were out of the specification range (Fig. 3.11). The radial load average varied from 16.5 oz/in. at 100% relative humidity to 22.6 oz/in. at 0% relative humidity. This variation requires 50% of the total specification range.

The humidity effect on hardness and lip opening pressure is more dramatic (Fig. 3.12). At 100% relative humidity, the average hardness is below specification, and half of the seals measured have hardness values below the low end of the specification. At 0% relative humidity, the average hardness has increased by 7.2 units. Variations in humidity result in hardness variations that consume 70% of the total hardness specification range.

The average value for lip opening pressure is above the upper specification limit at 0% relative humidity. Half of the seals measured were above specification. At 100% relative humidity, the LOP dropped 5.4 psi. Variations in humidity result in lip opening pressure variations that consume 75% of the LOP specification range.

Variations in humidity can result in large variations of key seal parameters used in quality control procedures. These variations can lead to disputes between customers and seal suppliers. It is recommended that polyacrylate seals be conditioned for 24 hr at 52% relative humidity before seal parameters are measured (Table 3.28).

Exposure to humidity does not affect functional performance. Seals made from polyacrylate materials were aged in environments with controlled humidity levels. The physical characteristics of the seals were measured before and after functional testing. The seals were tested for 300 hr. No leakage was observed, and the seal lip wear path was independent of the initial humidity level (Table 3.29). The lip hardness before the functional test varied with humidity. At 100% relative humidity, the hardness

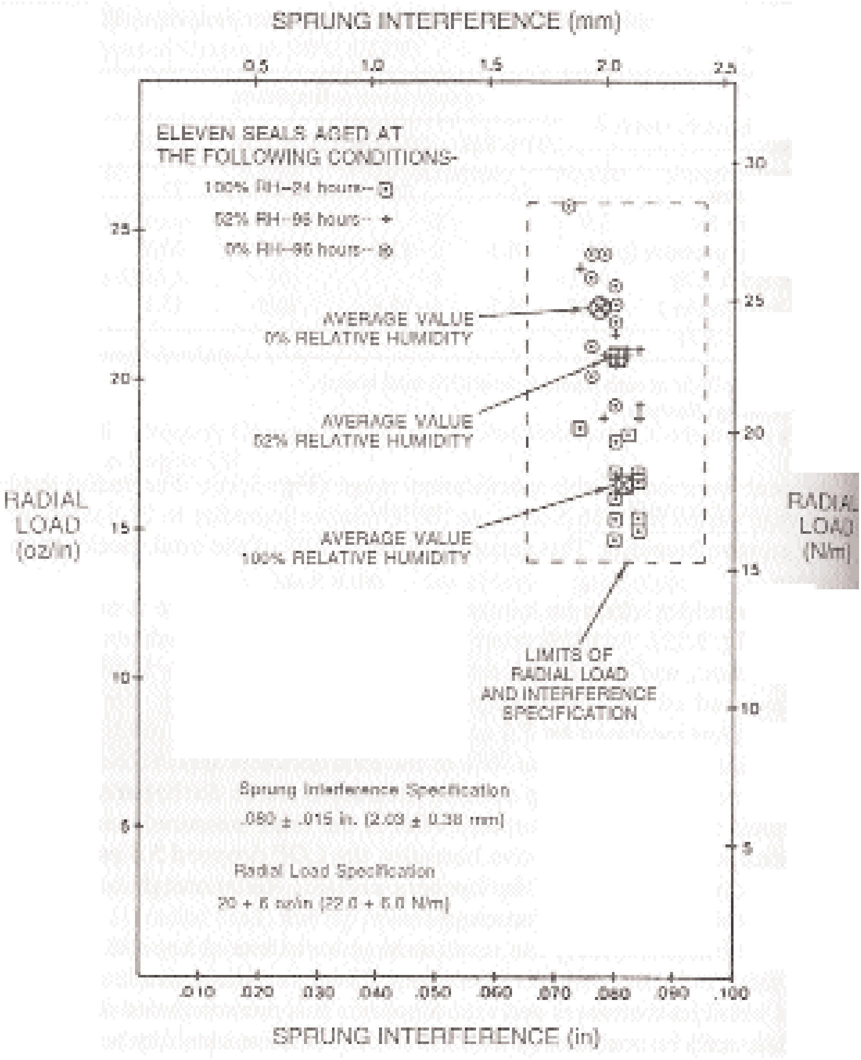


Fig. 3.11 Effect of humidity on polyacrylate quality control parameters (radial load and sprung interference). (Courtesy of Chicago Rawhide.)

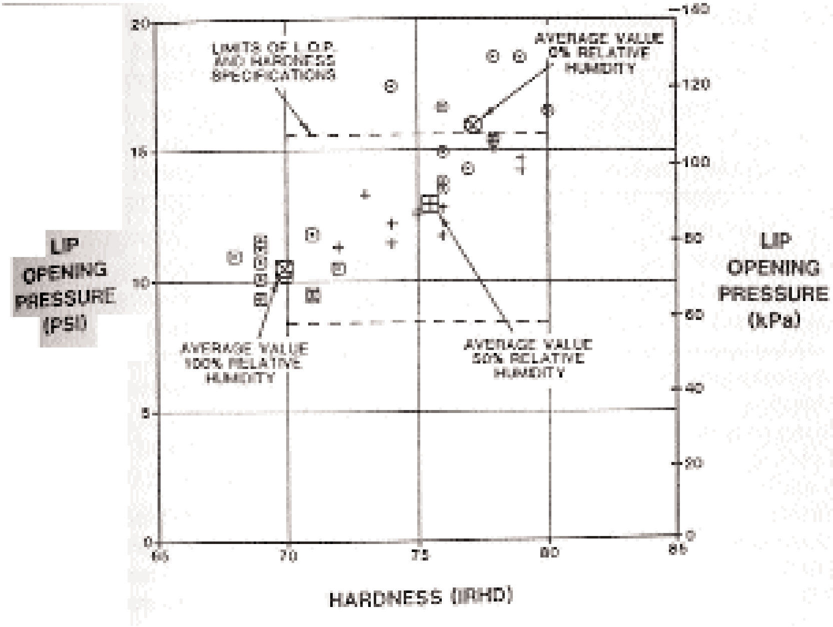


Fig. 3.12 Effect of humidity on polyacrylate quality control parameters (hardness and lip opening pressure). (Courtesy of Chicago Rawhide.)

Table 3.28 Solutions for Maintaining Constant Humidity[a]

Humidity level (%)	Ingredient
0	Dried calcium sulfate ($CaSO_4$)
20	Saturated aqueous solution of potassium acetate ($KC_2H_3O_2$)
52	Saturated aqueous solution of sodium dichromate ($Na_2Cr_2O_7$)
79	Saturated aqueous solution of ammonium chloride (NH_4Cl)
100	Distilled water

[a] Mixtures of water and certain solid chemicals will maintain the humidity of a closed chamber at a constant level. Different solids will result in different humidity levels. Enough excess solids must be provided that they are not completely dissolved in water.

Source: Chicago Rawhide.

Table 3.29 Functional Tests of Polyacrylate Seals Exposed to Various Levels of Humidity

Relative humidity (%)	Lip opening pressure (psi)		Radial load (oz/in.)		Hardness		Seal wear width (in.)
	Before	After	Before	After	Before	After	
0	10.5	5.0	16.6	9.7	73	76	0.010--0.040
20	10.5	5.5	17.0	9.3	72	75	0.017--0.038
52	10.4	5.5	16.8	9.1	71	76	0.015--0.035
79	9.4	5.5	16.6	9.0	67	77	0.014--0.030
100	8.0	5.0	13.9	9.1	58	74	0.016--0.038

No leakage observed during test.

Duration	300 hr
Sump temperature	135° C (275° F)
Lubricant	SAE 30
Sump level	Centerline
Shaft size	76.2 mm (3.000 in.)
Shaft speed	2165 rpm (8.72 m/sec, 1700 fpm)
DRO	0.13 mm (0.010 in.)
STEM	0.13 mm (0.010 in.)
Cock	Zero
Pressure	Zero
Cycle	20 hr on, 4 off

Source: Chicago Rawhide.

Table 3.30 Effect of Humidity on Low-Temperature Properties of Polyacrylate Materials

Relative humidity (%)	Temperature of failure (° C)		
	Impact brittleness test (ASTM D2137)	T_{10} Values for Gehman test (ASTM D1053)	Functional break test, 76.2 mm (3.000 in.) shaft size
0	--27	--32	--40
52	--26	--29	--43
100	--26	--30	--43

Source: Chicago Rawhide.

was only 58 compared to 73 at 0° relative humidity. All seals hardened during the functional test, and the post-test hardness values were independent of the starting humidity level. It is concluded that the absorbed water content of the material has no effect on the final measurements. This conclusion is also valid for radial load and lip opening pressure. Humidity affects the before-test values, but the water is driven off during testing and the post-test values are independent of the initial humidity levels.

Cold-temperature tests were also performed on polyacrylate seals exposed to different humidity levels. The impact brittleness test (ASTM D2137) consists of recording the temperature at which breakage occurs when a sample of the material is submitted to a known impact force. The Gehman test (ASTM D1053) measures the increase in the material modulus as temperature decreases. The T10 value is defined as the temperature that results in a modulus 10 times that at room temperature. The functional cold break test is a test with seals. Seals are placed on a 3 in. shaft with an 0.020 in. (0.50 mm) dynamic runout and zero shaft-to-bore misalignment. The temperature is lowered until breakage occurs. The humidity level has no effect on the results of the low-temperature tests (Table 3.30).

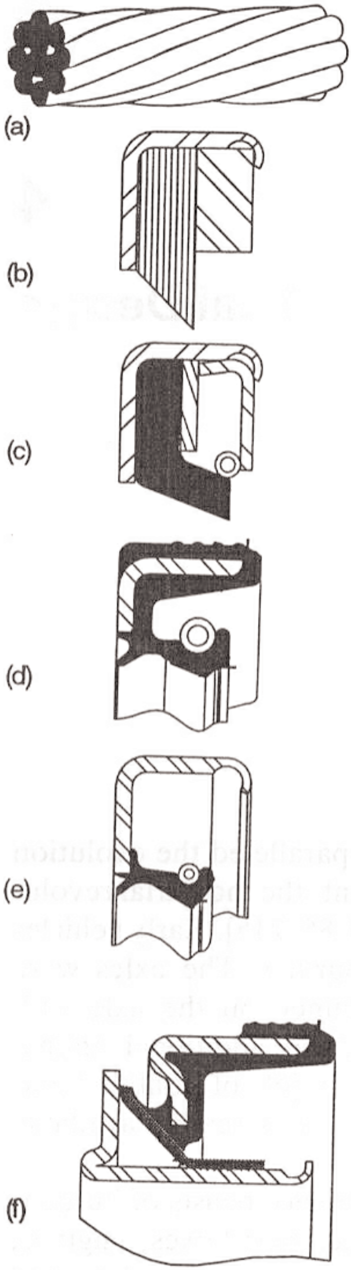
4

Seal Design

4.1 ROTARY SHAFT SEAL EVOLUTION

The evolution of rotary shaft seal technology has paralleled the evolution of the transportation industry, roadway development, the industrial revolution, and lubricants for moving components [80, 84, 85, 215]. Early vehicles were slow-moving wagons drawn by beasts of burden. The axles were lubricated with animal fat that was loosely contained on the axle with leather strips. The vehicle user periodically slapped additional animal grease onto the axle because the seals did a poor job of containment. Leakage was of little concern because the material was readily available and biodegradable.

The industrial revolution created machines to replace beasts of burden. Crude, heavy lubricants were developed to protect gear boxes, engines, and transmissions. The operating speeds and temperatures were low, and the lubricants did not degrade significantly. Early vehicles also used heavy greases that were adequate at the time. Roads were poor, and high vehicle speeds were impossible. Most seals of this era were made of organic fibers (flax, cotton, hemp) or leather. They were formed into a rope (Fig. 4.1a) configuration and wrapped around rotating, reciprocating, and oscillating shafts. These seals are known as packings, and they are still made today (but with modern materials) and are used in many industrial applications.


(a)
(b)
(c)
(d)
(e)
(f)

At the beginning of the twentieth century, rapid advances in technology were underway. Highways improved, vehicle speeds increased, industrial machinery became more sophisticated with higher shaft speeds and temperatures, and lubricants were blended to improve their properties, increase the wear resistance of gears, and handle heavier loads. These lubricants were also more expensive, less viscous, more harmful to the environment, and harder to seal than earlier greases and heavy lubricants. A new seal concept was developed in the 1920s. Leather lip seal elements were treated with chemicals to provide good oil resistance, and the leather was clinched in metallic cases to provide for easy installation and removal (Fig. 4.1b). The seal was pressed in a bore with a metal-to-metal pressfit that prevented bore leaks. The leather lip contacted the shaft, which had a region ground to a roughness that was recommended by the seal supplier.

The lubricants sometimes caused the leather to swell, and a spring was often added to the assembly to provide a more uniform load to extend seal life. Synthetic rubber was developed in the 1930s and 1940s, and by the end of World War II oil-resistant polymers called nitrile or NBR were replacing the leather material in assembled seals (Fig. 4.1c). Experimentation began with PTFE material, and by the 1960s some assembled seals were using a PTFE wafer instead of leather or synthetic rubber. PTFE is resistant to most fluids, but it is expensive and difficult to work with; therefore the number of applications was limited [51, 210].

Methods of bonding rubber to metals and other materials were developed, and the direct bond seal appeared in the 1950s (Fig. 4.1d). New high-temperature oil-resistant polymers were also developed. Silicone and poly-

Fig. 4.1 Shaft seal evolution (seal cross sections). (a) Packings were used from the 18th century to the present. Early packings were made with organic materials and leather. (b) Leather radial lip seals were used in the 1920s and 1930s. The leather was treated to be oil-resistant. (c) Assembled seals were used in the 1930s-1960s. The leather material was replaced by synthetic rubber (NBR) in the 1950s, and PTFE was introduced in the 1960s. (d) Bonded synthetic rubber has been used since the 1950s. Silicone material and polyacrylate were introduced in the 1960s and fluoroelastomer in the 1970s. (e) Reduced bond areas have improved cementing systems and have been used since the 1960s. (f) Unitized, composite seals have been used since the early 1980s. Several materials (polyacrylate, bonded PTFE, etc.) have been used to enhance properties and the sealing surface is an integral seal part. (g) Value-added composite seals were introduced in the 1990s. Several components are combined with the seal to provide a value-added package. The example shown combines seal, filter, reinforcing insert, and excluder into one package. (Courtesy of Chicago Rawhide.)

acrylate polymers were introduced in the 1960s, and fluoroelastomer began to get wide use in the 1970s. Some of these materials were much more expensive than the standard nitrile polymer, and seals were developed to reduce material content. Better cementing systems helped reduce material content by reducing the area required to ensure a good bond (Fig. 4.1e). Smaller seal lips also reduced material content and helped reduce seal torque. In the 1980s, seal designs were developed that include the sealing surface as an integral part of the design [99, 109, 138]. The seal supplier assumes full responsibility for the seal lip and the sealing surface. The seal user no longer has to prepare a special running surface on the shaft. The seal user no longer has to prepare a special running surface on the shaft. Methods were also developed to bond PTFE to rubber, and metal and direct bonded PTFE seals began to replace assembled designs. Costs were reduced with these composite seals (Fig. 4.1f), and the leakage path through the clinch area was eliminated. A high-temperature version of nitrile (HNBR) was also developed in the 1980s that provides good performance at both low and high temperatures.

Seal design has evolved another step in the 1990s. Seal manufacturers have begun to assume design responsibility for the sealing system instead of just the seal. As a result, several components in the sealing area are being combined with the seal in some applications to give the user a simple value-added package with fewer components (Fig. 4.1g) [30, 124, 125].

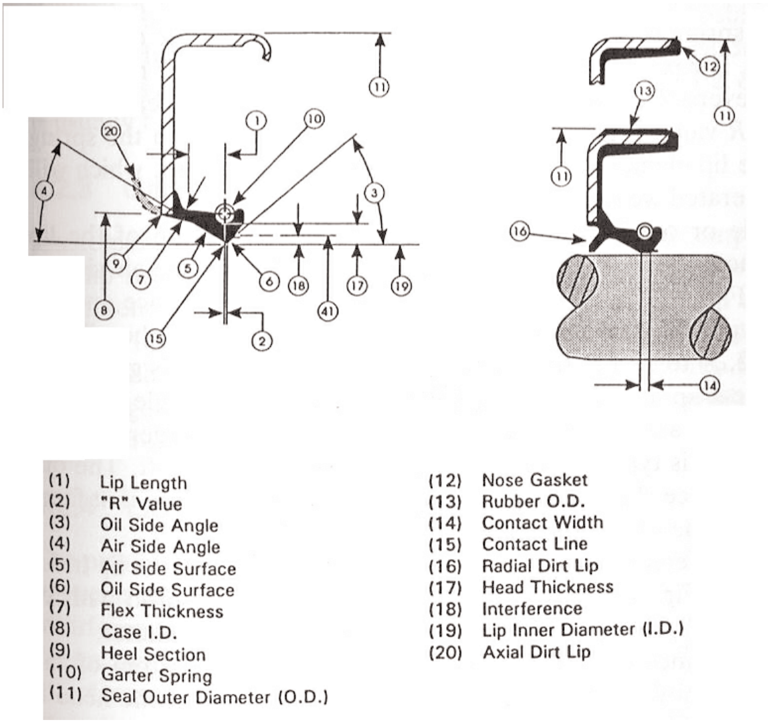
Future oil seals for dynamic applications will undoubtedly use new materials that are more oil- and temperature-resistant [86]. It is certain that the seal supplier, oil manufacturer, and seal user will work closely together to provide a value-added sealing system at a competitive cost.

4.2 SEAL ELEMENT GEOMETRY

The elastomeric material used to form the seal lip is one of the most important variables in the function and ultimate life of a seal. A well-designed seal lip section cannot compensate for the wrong base polymer selection or a poorly formulated compound. Seal design variables can also affect function and ultimate life. The seal lip is molded into a conical beam shape that has an inner diameter smaller than the shaft's outer diameter. During installation, the elastomeric lip is stretched over the shaft and a force is generated between the shaft and the lip contact area. This force consists of two components, a beam force and a stretch or hoop force. These forces are often complemented with a spring that will help compensate for changes that occur when the elastomer is exposed to fluids at elevated temperatures. The seal lip design must be selected so the forces will maintain a dam when the shaft is rotating, oscillating, or reciprocating at high speeds with vibration and dynamic runout. If the dam is broken, gaps between the seal lip will develop and gross leakage will result. The seal supplier and user must understand how seal performance varies when the design parameters that form a seal cross section are varied [78, 217].

4.2.1 Seal Cross Section

A typical seal cross section is diagrammed in Fig. 4.2; the dimensions will typically increase proportionally in size as shaft size increases [219]. The seal lip or beam length (Fig. 4.2, 1) is the axial distance from the flex point of the beam to the lip contact point [7, 189]. Increasing the beam length will increase flexibility, reduce beam forces, and improve the seal's ability



20
1
10
11
4
3
9
7
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18
17
19
8
41
15
2
12
13
11
11
16
14
(1) Lip Length
(2) "R" Value
(3) Oil Side Angle
(4) Air Side Angle
(5) Air Side Surface
(6) Oil Side Surface
(7) Flex Thickness
(8) Case I.D.
(9) Heel Section
(10) Garter Spring
(11) Seal Outer Diameter (O.D.)
(12) Nose Gasket
(13) Rubber O.D.
(14) Contact Width
(15) Contact Line
(16) Radial Dirt Lip
(17) Head Thickness
(18) Interference
(19) Lip Inner Diameter (I.D.)
(20) Axial Dirt Lip

Fig. 4.2 Sealing element geometry. (Courtesy of Chicago Rawhide.)

to follow shaft vibrations and dynamic runout. Decreasing the beam length will increase radial load and increase seal and shaft wear. Short, stubby lips are well suited for pressure applications. The beam length is normally between 2.5 and 5.0 mm (0.100--0.200 in.).

The flex thickness (Fig. 4.2, 7) is the thickness of the region that deforms when the seal is installed on the shaft. The flex thickness is normally 0.63--1.5 mm (0.024--0.060 in.) depending upon application and shaft size. Thick flex sections will increase the seal load on the shaft and reduce followability. Thick sections are often used in pressure and reciprocating applications. Thin flex sections reduce the seal load on the shaft and increase followability. If the section is too thin, the section may tend to "wrap" around a rotating shaft, which may result in "slip--stick". The heel section (Fig. 4.2, 9) of the lip is the region where the elastomer is bonded to the case.

The R value (Fig. 4.2, 2) is the axial distance between the spring centerline and the lip contact point and should have a value between 0.35 and 0.75 mm (0.010--0.030 in.) when the seal is installed on the shaft. The spring centerline must be positioned toward the air side of the seal contact point. This condition is known as a positive R value. If the R value is negative, the spring centerline is positioned toward the oil side of the seal contact point. The pressure distribution between the lip contact region and the shaft is reversed if the R value is negative, and instant leakage will result. If the R value is greater than 0.75 mm (0.030 in.), then the spring may cause the lip element to collapse or develop a bell mouth, which will result in accelerated wear and short life.

The air-side or outside surface (Fig. 4.2, 5) is the portion of the lip surface that faces the air side or outside of the application. The air-side or barrel angle (Fig. 4.2, 4) is the angle between the air-side surface and the shaft and is typically 20--35° on shaft. It must be smaller than the oil-side angle (Fig. 4.2, 3) to ensure that the desired pressure gradient is generated between the seal lip and the shaft. The oil-side angle is the angle between the oil-side surface and the shaft surface. This angle must be larger than the air-side angle and is typically 40--70"° when installed on the shaft. The oil-side or inside surface (Fig. 4.2, 6) is the portion of the lip surface that faces the oil side or inside of the application.

The inside diameter of the primary lip is typically known as the lip inner diameter (I.D.) (Fig. 4.2, 19). The dimension can be measured with or without the garter spring and will be smaller than the shaft diameter.

The radial distance between the contact point and the bottom of the spring groove is known as the head thickness (Fig. 4.2, 17). If the head is too thin, instability will result, and the element may lift off the shaft, resulting in leakage. The on-shaft oil- and air-side angles may also fluctu1-

ate, creating leakage. Slip-stick may result. Head thickness is typically 1 25--2.5 mm (0.050--0.100 in.) and increases with shaft size.

4.2.2 Seal Case

The elastomeric member is bonded to a rigid member, which is usually steel and is called the seal case. The outer diameter (O.D.) (Fig. 4.2, 11) of the seal assembly is larger than the housing bore. When installed, the pressfit that results prevents leakage. In some designs, the elastomer is extended over the face of the case to form a nose gasket (Fig. 4.2, 12). A gasket is formed when the seal is installed and bottomed in a stepped housing bore. Nose gaskets are used to prevent bore leakage if the bore conditions and dimension do not meet recommended specifications. The metal portion of the seal will still have a pressfit with the bore to prevent the seal from backing out of the bore. In other designs, the rubber flows completely across the case width, and a full rubber O.D. (Fig. 4.2, 13) is formed. Rubber O.D. seals are used to prevent leakage in a through bore where the seal cannot be bottomed and the bore conditions and dimensions do not meet recommended specifications. Rubber O.D. seals are often used in nonferrous housings that have a larger coefficient of thermal expansion [119]. The smallest diameter of the seal case is known as the case inner diameter (I.D.) (Fig. 4.2, 8). The seal lip is usually bonded to the case in this region.

4.2.3 Garter Spring

Elastomers will absorb oil when exposed to elevated temperatures. The seal lip softens and tends to expand, which results in a reduction of lip interference and load between the seal lip and the shaft. In extreme cases, the seal lip can expand completely off the shaft, and gross leakage will result. A helically coiled spring with the ends connected to form a circular ring or garter spring (Fig. 4.2, 10) is often used to minimize load and interference loss to extend the life of the seal.

4.2.4 Lip/Shaft Contact Region

The primary seal lip diameter is less than the shaft diameter. The difference between these diameters when the seal is sprung is known as the sprung interference (Fig. 4.2, 18). The unsprung interference is the difference between the unsprung seal lip diameter and the shaft diameter. Excessive interference will generate a high radial load, high torque, and excessive underlip temperature. If the interference is too small, then the element will not follow shaft vibrations and dynamic runout. Gross leakage may result without any evidence of seal lip or shaft wear. The interference is dependent upon the seal lip material, the application conditions, and the shaft size. Values for interference can range from 1.25 to 3.75 mm (0.050--0.150 in.).

The contact line (Fig. 4.2, 15) in the lip/shaft contact region is the circular line that is formed where the air-side and oil-side surfaces of the elastomeric lip intersect. The contact line is seen as a point when the seal element is viewed in a cross section. When the seal is installed on the shaft, the resulting forces deform the sealing tip, which results in a contact region with a finite width (Fig. 4.2, 14) in the axial direction. This width is usually about 0.10 mm (0.005 in.) when the seal is new and may be as much as 2.5 mm (0.100 in.) when the seal is worn.

4.2.5 Contaminant Exclusion Lips

The radial dirt lip (Fig. 4.2, 16) is a short, nonsprung lip on the seal air side that extends toward the shaft in the radial direction and is used to exclude contaminants. The design is not as effective as the axial dirt lip (Fig. 4.2, 20) [79, 139, 140]. This nonsprung lip extends in the axial direction on the air side of the seal and impinges upon a flange or other surface that typically extends radially outward from the shaft surface. Two opposed sprung lips are sometimes used to enhance contaminant exclusion.

4.3 SEAL TYPES

4.3.1 Springless Seals

Some low-cost radial lip seals have no springs (Fig. 4.3a) and are used to seal grease at low speeds and low temperatures. Shaft speeds are usually less than 10 m/sec (2000 ft/min), and the temperature is under 100˚ C (212˚ F).

In some applications, the primary lip faces the air side to exclude contaminants. If the contamination is heavy, the inside of the case can be covered with rubber to eliminate corrosion (Fig. 4.3b). Stainless steel cases can also be used, but the seals will be more expensive than seals made with mild steel cases. Double-lip seals will prevent grease leakage and exclude contaminants (Fig. 4.3c). Seals of this type are used in vehicle wheels, conveyor rollers, grease-packed gear boxes, and many other grease-lubricated applications. Seal designs can be specialized to include more than two lips. Some seals have an axial dirt lip, which has been shown to be the best contaminant excluder lip (Fig. 4.3d). These are used in wheel applications that are sealed and greased for the life of the vehicle. Other heavy duty multilip seals (e.g., Fig. 4.3e) have grease packed between the lips and are used in very dirty applications where speeds are less than 2.5 m/sec (500 ft/min). These seals are used in farm equipment.

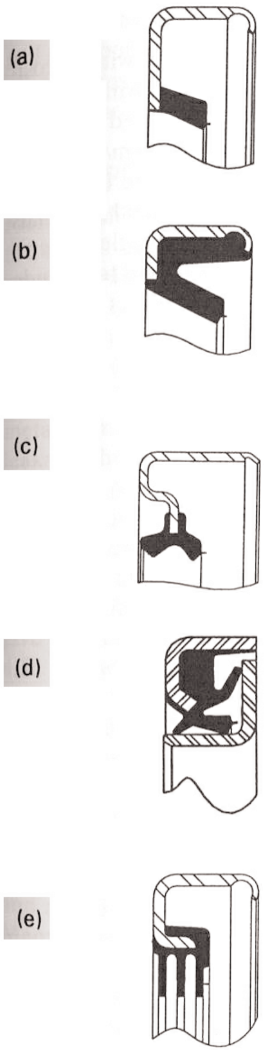


Fig. 4.3 Springless seals. (a) Grease seal for temperatures 100˚ C (212˚ F) and shaft speeds 10 m/sec (2000 ft/min). (b) Excluder seal, where the lip faces the air side with rubber on the inside face to prevent corrosion. (c) A dual-lip seal seals grease and excludes contaminants. (d) The multilip seal, where an axial dirt lip excludes contaminants and a stainless steel sleeve resists corrosion. (e) Heavy duty multilip seals are grease-packed to resist heavy contaminants. To be used with shaft speeds 2.5 m/sec (500 ft/min). (Courtesy of Chicago Rawhide.)

4.3.2 Spring-Loaded Seals

Springs are added to seal lips when the sealed lubricants are oils with viscosities lower than that of grease. The spring helps provide a uniform load and resists the elastomer swell when it is exposed to the oil at elevated temperatures. Shaft speeds for standard seal designs are extended to 25 m/sec (5000 ft/min). Maximum application temperatures are limited by the elastomer used to mold the sealing lip (see Chapter 2). Special designs and materials can extend the upper speed limit to 50 m/sec (10000 ft/min). Single-lip seals (Fig. 4.4a) are the most common designs. A radial dirt lip is used to exclude light contaminants in applications where the seal area is protected (Fig.

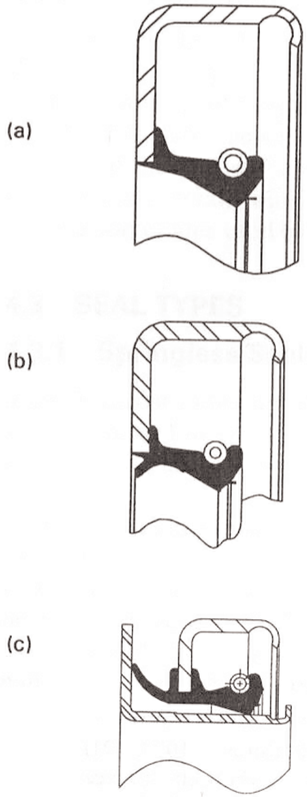


Fig. 4.4 Spring-loaded seals, (a) Single lip for shaft speeds 25 m/sec (5000 ft/min). (b) Dual-lip seal has a radial dirt lip that excludes light dust. (c) A dual-lip seal with an axial dirt lip excludes heavy contaminants.

4.4b). The axial dirt lip (Fig. 4.4c) is recommended if the seal is unprotected and the contamination is heavy. Spring-loaded seals are used in a variety of applications that range from slow-moving wheels to high-speed, high-temperature engines.

4.3.3 Special Seals

Many applications have unique requirements that cannot be satisfied with the conventional seal designs shown in Figs. 4.3 and 4.4. A multitude of special designs exist to handle the requirements of the special applications. Some applications require materials that resist harsh chemicals, sparse lubrication, and high temperatures. If elastomers cannot withstand the application conditions, then PTFE wafers are often used [15, 210]. PTFE is difficult to bond to other materials, and the wafer can be clamped between metal components to form a seal (Fig. 4.5a). A rubber gasket prevents leakage through the clinch area. Radial lip seals are also used in high-pressure applications. A plastic backup ring prevents lip inversion during pressure surges. The seal in Fig. 4.5b is used in reciprocating applications with pressure spikes up to 10,000 kPa (1500 psi). Similar designs can be used in rotating and oscillating applications. Some applications may have excessive shaft vibrations or dynamic runout. Seal lips can be convoluted to give greater followability in the radial direction. The design shown in Fig. 4.5c can withstand dynamic runouts up to 1.5 mm (0.060 in.) for short periods of time without leakage. Conventional lips may leak instantly if the dynamic runout is 0.5 mm (0.020 in.).

Seals are often combined with other components to form a value-added package. Figure 4.5d combines the high-pressure seal design of 4.5b with an excluder lip on the air side and a filter on the oil side [125]. The filter strains the oil and prevents internal contaminants such as core sand from damaging the seal lip. A dual-lip seal is combined with a plastic retainer and a gasket to form the value-added package shown in Fig. 4.5e [124]. Many other special seals can be designed to handle unique application conditions that exceed the capabilities of conventional seals. The seal user should consult with the seal supplier to obtain a solution for the most difficult conditions.

4.3.4 Unitized Seals

The radial lip seal is actually one-third of the sealing system. The seal user must provide a well-designed bore and shaft that meet specifications (see Chapter 5). If the shaft does not meet specifications, the seal can be unitized with a running surface as part of the seal package. The seal shown in Fig. 4.6a is designed for a stationary spindle and a rotating hub. The seal

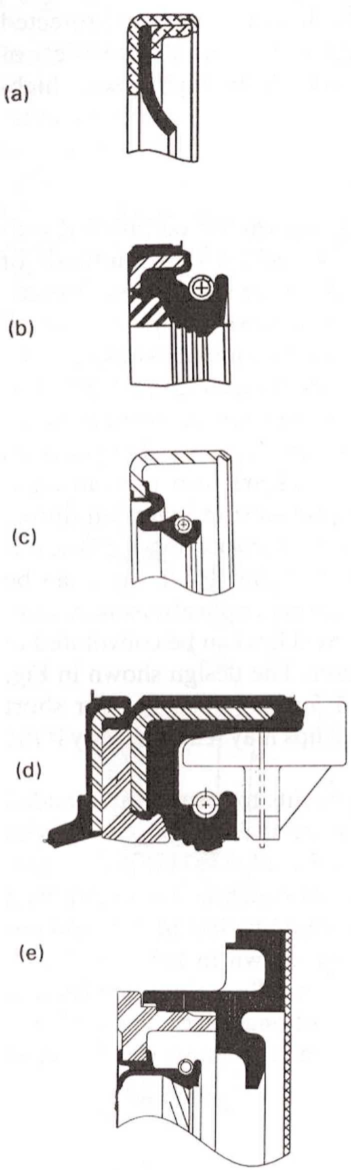


Fig. 4.5 Special seal designs. (a) An assembled seal for materials that are difficult to bond. (b) A high-pressure seal with backup ring. (c) A convoluted lip is used for high-runout applications. (d) The value-added seal has a filter on the oil side and an excluder lip on the air side. (e) This value-added seal combines seal, seal retainer, and gasket. (Courtesy of Chicago Rawhide.)

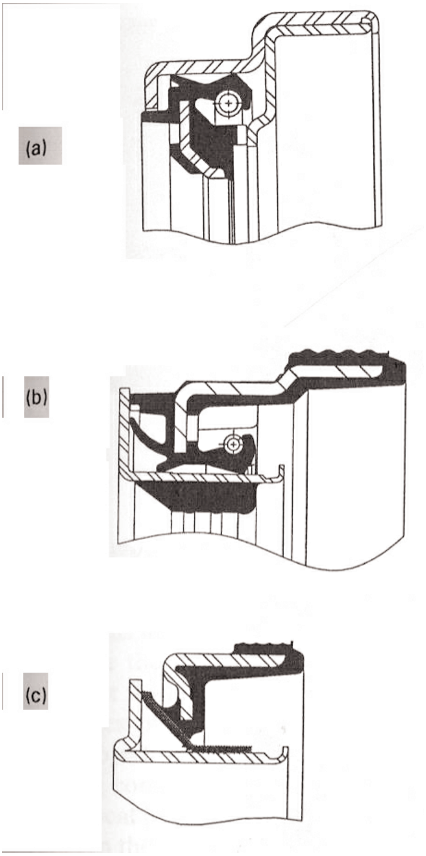


Fig. 4.6 Unitized seals. (a) An external seal for rotating hubs. (b) The rubber I.D. and O.D. for hand installation provide an axial dirt lip to exclude heavy contaminants. (c) A composite seal with PTFE wafer bonded to rubber provides an axial dirt lip to exclude heavy contaminants. (Courtesy of Chicago Rawhide.)

lip is stationary and seals against the rotating outer stamping, which is part of the unitized package. The unitized seal shown in Fig. 4.6b has a rubber I.D. and O.D., which allows hand installation for replacement in the field [138]. It also features an axial dirt lip to exclude heavy contaminants.

Several materials can be used to fabricate a seal. A PTFE wafer is bonded to rubber, which is bonded to the metal case. The seal is unitized with a metal sleeve that provides the running surface for the primary lip and a vertical flange for the axial PTFE excluder lip (Fig. 4.6c) [99]. Unitized seals relieve the user from providing a shaft with a ground surface. The seal supplier assumes the responsibility for the sealing system.

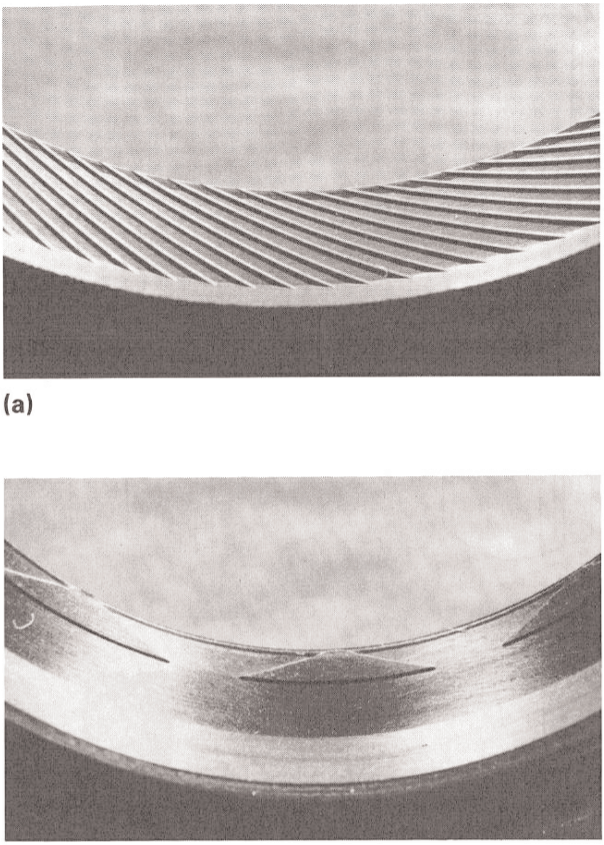


Fig. 4.7 Hydrodynamic seals. (a) Helix seal. (b) Birotational seal with triangular pads. (c) Birotational seal-sine.

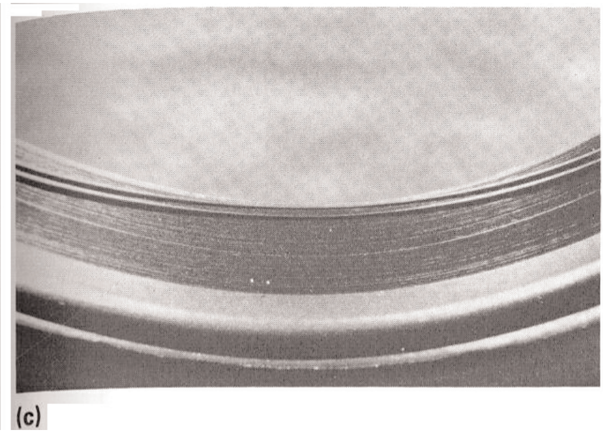


Fig. 4.7 Continued

4.3.5 Hydrodynamic Seals

The sealing mechanism depends on a properly compounded seal material that forms microasperities in the seal lip wear track. These microasperities lubricate the wear track and also provide a pumping action that pushes lubricant toward the sump (see Chapter 6). Some seals use ribs, pads, or other design features to augment this microscopic pumping ability [48]. The projections are molded on the air side and have the disadvantage of pushing contaminants toward the seal/shaft contact region, which may increase seal lip and shaft wear. High pump rates can also generate a vacuum between the primary lip and a secondary exclusion lip, which can cause the lip to collapse on the shaft, which in turn results in leakage. Seals with molded pumping aids are sometimes called hydrodynamic seals.

The helix seal (Fig. 4.7a) has ribs molded on the air side that form an angle of about 20° with the contact line between the lip and the shaft. The ribs are blended into a static band to ensure that no gaps exist in the contact zone. The helix seal can be used only with an application that has shaft rotation in one direction (unidirectional). If the direction of shaft rotation is reversed, gross leakage will result because the ribs will pump fluid out of the sump. A screw thread is sometimes molded or machined into the seal material on the air side. This seal is called a spiral seal and is also unidirectional. A birotational seal can be developed by molding triangular pads onto the air side of the seal (Fig. 4.7b) [223]. The sides of the pads create helices that will pump oil into the sump regardless of the direction of shaft rotation. The sine seal (Fig. 4.7c) has ribs molded on the air side in the

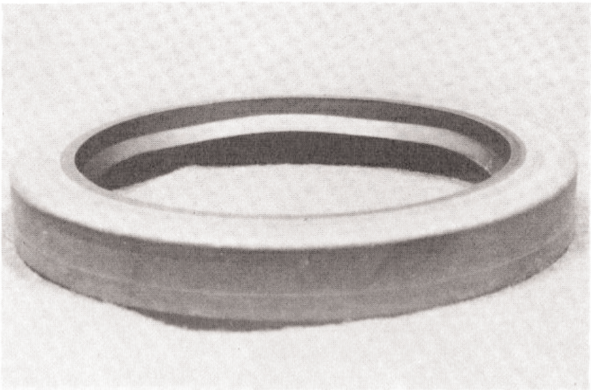


Fig. 4.8 Wave seal has no ribs but provides a pump rate in both directions of shaft rotation.

shape of a sine wave. This design is also a birotational seal that pumps in both directions of shaft rotation. The wave seal (Fig. 4.8) has no ribs that can pump contaminants into the sump. The entire lip is molded into a wave that forms a sinusoidal pattern when the seal is installed on the shaft. This wave pattern provides a moderate pumping action in both directions of shaft rotation, increases the heat transfer area on the shaft, lowers underlip temperature, and minimizes contaminant ingestion [69, 70, 74]. The pump rate can be increased by increasing the wave amplitude and the number of wave cycles. Helix and spiral seals have the highest pump rates, the wave seal has a moderate pump rate, the triangle seal pump rate is lower, and the pump rate of a plain lip seal without pumping aids is the lowest (Fig. 4.9). The rib patterns can be examined by installing the seal on a transparent shaft (Fig. 4.10). Determination of a good contact pattern is subjective. Examples of good and bad patterns are shown in Fig. 4.11.

4.4 SEAL CASE

4.4.1 Materials

The elastomeric member is normally bonded to a rigid case. This case protects the seal lip, and its outer diameter provides a pressfit surface that fits into the housing. The case is normally formed from 1005 to 1020 cold-rolled steel. If the case is exposed to the elements, a zinc plating with a clear chromate or dichromate dip is sometimes applied. Cases may also be made from stainless steel, brass, or aluminum. These materials are more

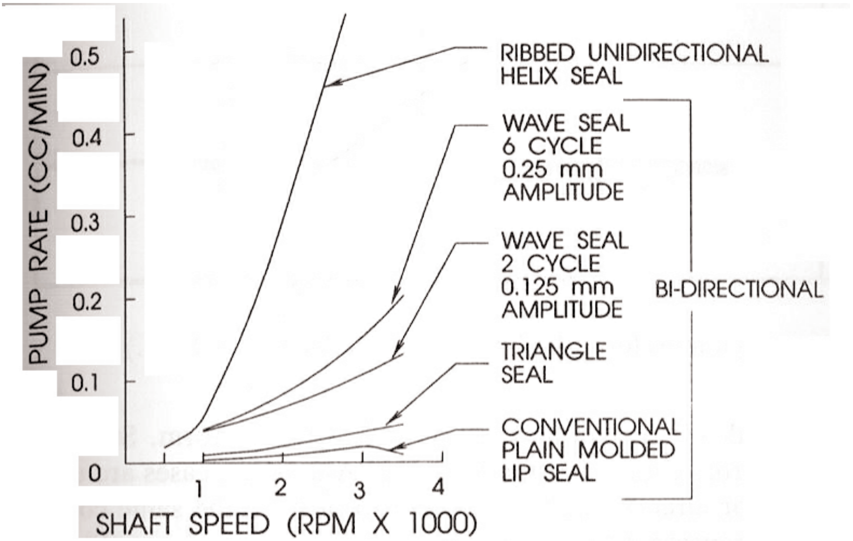


Fig. 4.9 Measured pump rates for hydrodynamic seals. (Courtesy of Chicago Rawhide.)

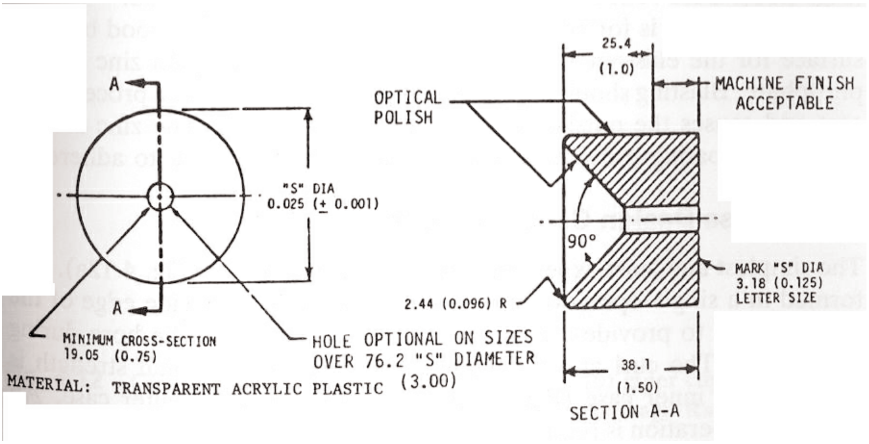

A
A
"S" DIA
0.025 (± 0.001)
MINIMUM CROSS-SECTION
19.05 (0.75)
HOLE OPTIONAL ON SIZES
OVER 76.2 "S" DIAMETER
(3.00)
MATERIAL: TRANSPARENT ACRYLIC PLASTIC
OPTICAL
POLISH
25.4
(1.0)
MACHINE FINISH
ACCEPTABLE
90°
2.44 (0.096) R
MARK "S" DIA
3.18 (0.125)
LETTER SIZE
38.1
(1.50)
SECTION A-A

Fig. 4.10 Transparent acrylic plastic shaft configuration to view seal contact Pattern. (Courtesy of SAE.)

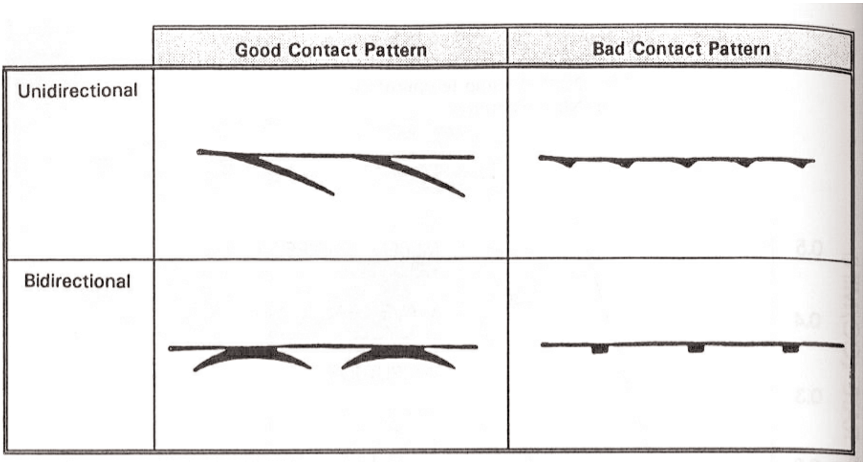


Fig. 4.11 Contact patterns for hydrodynamic seals. (Courtesy of SAE.)

expensive than cold-rolled steel and are more difficult to form. Stainless steel is used if external contamination is severe. Aluminum cases are used to reduce weight for aircraft applications and to provide the same coefficient of thermal expansion if the case is installed in aluminum bores. Plastic cases are usually not used because of their high coefficient of thermal expansion, poor thermal conductivity, and low modulus. Plastics are used to replace metal retainers that are gasketed and bolted to machine housings (Fig. 4.5e). Cold-rolled steel is the most common material used.

After the case is formed, it must be treated to provide a good bonding surface for the elastomer. It can be blasted or covered with zinc or iron phosphate. Blasting should be avoided if possible, because the process adds cost and causes the outside diameter to become larger. The zinc or iron phosphate coating provides a rough surface for the cement to adhere to.

4.4.2 Case Design Configurations

The simplest and least expensive design is the L cup case (Fig. 4.12a). It is formed in a single operation in a stamping press. The outside edge of the cup is curled to provide a chamfer to guide the seal into the bore during installation. The curl also strengthens the case. If additional strength is required, an inner case (Fig. 4.12b) is inserted into the outer case. An additional operation is required to curl the outer case over the inner case to hold it tightly in place. The inner case also protects the lip during handling and shipping and helps prevent the spring from popping out of the spring groove during handling. The stepped case (Fig. 4.12c) also increases case

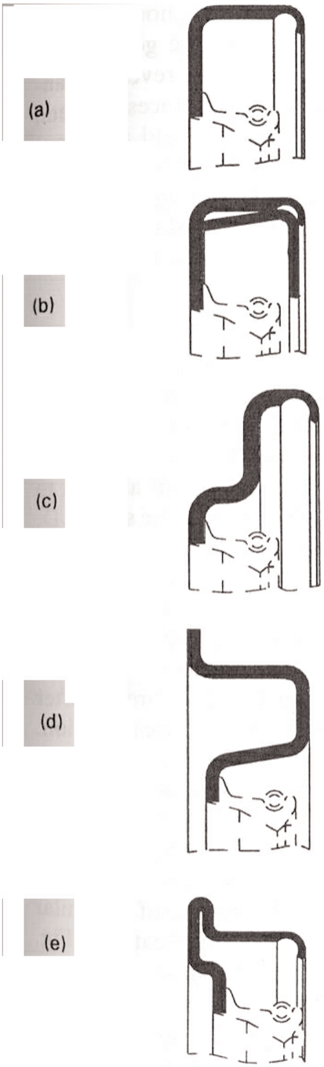


Fig. 4.12 Case design configuration. (a) L cup case; (b) inner cup; (c) stepped case; (d) reverse channel; (e) shotgun case. (Courtesy of Chicago Rawhide.)

strength and is used if there is large clearance between the housing bore and the shaft. It is used to offset the sealing lip if there are geometrical obstacles in the application that will create interference. The reverse channel case (Fig. 4.12d) is also used to accommodate large clearances between the housing bore and the shaft. It increases case strength, and a pry-out flange can be added to assist in seal removal. A shotgun case (Fig. 4.12e) also has a pry-out flange and increases case strength. This design should be avoided because it is expensive to manufacture and unwanted drawing oils can be trapped in the gap created between the metals in the flange area.

4.4.3 Seal Case Gaskets

If the bore is made of steel and meets recommended specifications (see Chapter 5, Section 5.3), then a seal with a properly designed steel O.D. (Section 5.4.1) will not leak after installation. Rough bore finishes, axial scratches, corrosion, dissimilar bore and case materials, bore or seal dimensions not meeting specifications, and damaged seal O.D.s can all lead to leakage between the seal case and the bore after installation. The seal O.D. design can be altered to help prevent bore leakage.

The simplest approach is to coat the metal seal O.D. with a thin sealant layer (Fig. 4.13a). This layer can be applied by the seal manufacturer. If an O.D. sealant is applied by the seal user in the field, care must be exercised to ensure that the material does not drip on the seal lip and reduce the flexibility. The O.D. sealant acts as a lubricant to reduce installation forces. After installation, the sealant will flow into the bore pores and increase retention. O.D. sealants are not recommended if the bore exceeds roughness specifications or has scratches deeper than 0.025 mm (0.001 in.). If the seal diameter is smaller than 25 mm (1.000 in.), then an O.D. sealant will prevent leakage when the coefficient of thermal expansion of the housing is larger than that of the seal case material. O.D. sealants are not recommended to prevent leakage for large seals when the seal case and housing are made of dissimilar materials. A rubber O.D. seal is recommended for these applications (Fig. 4.13b). If rubber O.D. seals are used in high-pressure applications, then a retaining ring such as a snap ring is recommended.

If the bottom of the housing bore is not finished to proper specifications, then a nose gasket (Fig. 4.13c) is often used to prevent leakage. The rubber coating on the oil side of the seal will also prevent case corrosion from the sealed media. A heel gasket (Fig. 4.13d) provides a rubber coating on the outside of the case to prevent corrosion from the external contaminants. The rubber portion of the O.D. provides a gasket to prevent bore leakage. The steel portion of the O.D. provides a lead-in chamfer during installation and a metal-to-metal pressfit that provides adequate retention forces [46, 100].

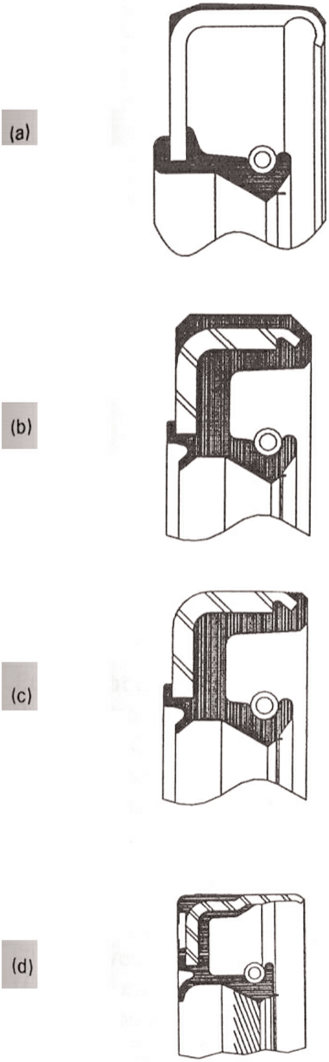


Fig. 4.13 Seal case gaskets. (a) O.D. sealant; (b) rubber O.D.; (c) nose gasket; (d) heel gasket. (Courtesy of Chicago Rawhide.)

5

Specifications and Recommended Practices

5.1 SEAL ENVELOPE DIMENSIONS

Equipment designers must provide adequate room for the seal. Many times the seal requirements are forgotten until after design work is complete. Expensive design changes may be required or a compromise may be required in seal selection that will result in poor sealing performance. Guides for seal design and specifications have been published by the Rubber Manufacturers Association [153], the Society of Automotive Engineers [195], and others.

The seal envelope is defined by specifying the seal width (Ws) and the radial cross section (Rcs). The radial cross section is one-half of the difference between the nominal bore diameter and the nominal shaft diameter (Ds). The envelope increases as the shaft diameter increases. The recommended nominal dimensions in inches for the seal envelope appear in Table 5.1 Table 5.2 contains the metric dimensional recommendations for the seal envelope. If the bore is recessed into the housing, the required bore Width (Wb) for a given seal width can be determined with Equation (5.1) for inch dimensions and Equations (5.2) and (5.3) for metric dimensions.

$$\mathrm{Wb} = \mathrm{Ws} + 0.016 \text{ in.} \tag{5.1}$$

$$\mathrm{Wb} = \mathrm{Ws} + 0.9 \text{ mm } (\mathrm{Ws} \le 10) \tag{5.2}$$

$$\mathrm{Wb} = \mathrm{Ws} + 1.2 \text{ mm } (\mathrm{Ws} > 10) \tag{5.3}$$

Table 5.1 Radial Lip Seal Envelope (dimensions in inches)

D_B	R_S	R_{CS}	W_S	D_B	R_S	R_{CS}	W_S	D_B	R_S	R_{CS}	W_S
.999	.500	A	B	2.750	1.750	E	J	4.003	3.000	E	K
1.124	.500	B	B	2.623	1.875	C	J	4.125	3.000	F	K
1.250	.500	C	B	2.750	1.875	D	J	4.125	3.125	E	E
1.124	.625	A	B	2.875	1.875	E	J	4.249	3.125	F	E
1.250	.625	B	B	3.000	1.875	F	J	4.376	3.125	G	E
1.375	.625	C	B	3.125	1.875	G	J	4.500	3.125	H	E
1.499	.625	D	B	2.623	2.000	B	J	4.249	3.250	E	E
1.250	.750	E	B	2.750	2.000	C	J	4.376	3.250	F	E
1.375	.750	B	B	2.875	2.000	D	J	4.500	3.250	G	E
1.499	.750	C	B	3.000	2.000	E	J	4.626	3.250	H	E
1.624	.750	D	B	3.125	2.000	F	J	4.249	3.375	D	E
1.375	.875	A	B	2.750	2.125	B	K	4.376	3.375	E	E
1.499	.875	B	B	2.875	2.125	C	K	4.500	3.375	F	E
1.624	.875	C	B	3.000	2.125	D	K	4.626	3.375	G	E
1.752	.875	D	B	3.125	2.125	E	K	4.376	3.500	D	E
1.499	1.000	A	B	3.251	2.125	F	K	4.500	3.500	E	E
1.624	1.000	B	B	3.000	2.250	C	K	4.626	3.500	F	E
1.752	1.000	C	B	3.125	2.250	D	K	4.751	3.500	G	E
1.874	1.000	D	B	3.251	2.250	E	K	4.626	3.625	E	E
1.624	1.125	A	B	3.371	2.250	F	K	4.751	3.625	F	E
1.752	1.125	B	B	3.125	2.375	C	K	4.876	3.625	G	E
1.874	1.125	C	B	3.251	2.375	D	K	4.999	3.625	H	E
2.000	1.125	D	B	3.371	2.375	E	K	4.626	3.750	D	E
1.752	1.250	A	B	3.500	2.375	F	K	4.751	3.750	E	E
1.874	1.250	B	B	3.251	2.500	C	K	4.876	3.750	F	E
2.000	1.250	C	B	3.371	2.500	D	K	4.999	3.750	G	E
2.125	1.250	D	B	3.500	2.500	E	K	4.876	3.875	E	E
2.000	1.375	B	J	3.623	2.500	F	K	4.999	3.875	F	E
2.125	1.375	C	J	3.371	2.625	C	K	5.125	3.875	G	E
2.250	1.375	D	J	3.500	2.625	D	K	5.251	3.875	H	E
2.374	1.375	E	J	3.623	2.625	E	K	4.999	4.000	E	E
2.125	1.500	B	J	3.751	2.625	F	K	5.125	4.000	F	E
2.250	1.500	C	J	3.500	2.750	C	K	5.251	4.000	G	E
2.374	1.500	D	J	3.623	2.750	D	K	5.375	4.000	H	E
2.502	1.500	E	J	3.751	2.750	E	K	5.251	4.250	E	E
2.250	1.625	B	J	3.875	2.750	F	K	5.375	4.250	F	E
2.374	1.625	C	J	3.623	2.875	C	K	5.501	4.250	G	E
2.502	1.625	D	J	3.751	2.875	D	K	5.625	4.250	H	E
2.623	1.625	E	J	3.875	2.875	E	K	5.501	4.500	E	E
2.374	1.750	B	J	4.003	2.875	F	K	5.625	4.500	F	E
2.502	1.750	C	J	3.751	3.000	C	K	5.751	4.500	G	E
2.623	1.750	D	J	3.875	3.000	D	K	5.751	4.750	E	F

Table 5.1 (Continued)

D_B	R_S	R_{CS}	W_S	D_B	R_S	R_{CS}	W_S	D_B	R_S	R_{CS}	W_S
6.000	4.750	G	F	6.500	5.250	G	F	6.750	5.750	E	F
6.000	5.000	E	F	6.625	5.250	H	F	6.875	5.750	F	F
6.250	5.000	G	F	6.500	5.500	H	F	7.000	5.750	G	F
6.375	5.000	H	F	6.625	5.500	F	F	7.125	5.750	H	F
6.250	5.250	E	F	6.750	5.500	G	F	7.125	6.000	F	F
6.375	5.250	F	F	6.875	5.500	H	F	7.500	6.000	I	F

D_B, Bore diameter; D_S, shaft diameter; R_{CS}, radial cross section (approximate); W_S, seal width; W_B, bore width.

A 0.250 in.
B 0.312 in.
C 0.375 in.
D 0.437 in.
E 0.500 in.
F 0.563 in.
G 0.625 in.
H 0.687 in.
I 0.750 in.
J 0.375 in. without inner cup
J 0.500 in. with inner cup
K 0.437 in. without inner cup
K 0.500 in. with inner cup

$W_B = W_S + 0.016$ in. (for recessed bore)

Source: Chicago Rawhide.

5.2 SHAFT RECOMMENDATIONS AND SPECIFICATIONS

5.2.1 Shaft Material and Geometry

The radial lip seal is only one-third of the sealing system. The shaft geometry [216] and the proper preparation of the shaft surface that the seal lip contacts forms another essential third of the sealing system [98]. Mild steel, cast iron, or malleable iron shafts are typically used. The shaft surface that the seal lip contacts should be hardened to a minimum of Rockwell C-30. If handling damage is likely to occur, then additional hardening to Rockwell C-45 is recommended. Chrome or nickel plating can provide a hard surface and also prevent corrosion in harsh environments. Brass, bronze, aluminum, zinc, magnesium, and other soft materials should not be used because excessive shaft wear may occur. Hardened wear sleeves of mild steel should be pressed over the shaft if these materials are used.

Leakage between the wear sleeve and the shaft may occur because of differential thermal expansion. Slippage of the wear sleeve may also occur. This is more likely if the shaft O.D. is large. Sealants are sometimes placed on the inside of the wear sleeve to prevent leakage and lock the wear sleeve firmly on the shaft. Care must be taken to ensure that the sealant does not smear on the outside surface of the wear sleeve. This material could impair the sealing mechanism and cause leakage. Mild wear sleeves are also used with mild steel shafts to avoid having to replace the entire shaft when wear in the seal contact area is generated. The damaged wear sleeve is removed, and a new one is substituted. The wear sleeve should be replaced whenever

Table 5.2 Radial Lip Seal Envelope (Dimensions in mm)

D_B	D_s	R_{CS}	W_s	D_B	D_s	R_{CS}	W_s	D_B	D_s	R_{CS}	W_s
16	6	5	7	42	30	6	7	95	75	10	10
22	6	8	7	47	30	8.5	7	100	75	12.5	10
22	7	7.5	7	52	30	6	7	100	80	10	10
22	8	7	7	45	32	6.5	8	110	80	15	10
24	8	8	7	47	32	7.5	8	110	85	12.5	12
22	9	6.5	7	52	32	10	8	120	85	17.5	12
22	10	6	7	50	35	7.5	8	120	90	15	12
25	10	7.5	7	52	35	8.5	8	120	95	12.5	12
24	12	6	7	55	35	10	8	125	100	12.5	12
25	12	6.5	7	55	38	8.5	8	140	110	15	12
30	12	9	7	58	38	10	8	150	120	15	12
26	15	5.5	7	62	38	12	8	160	130	15	12
30	15	7.5	7	55	40	7.5	8	170	140	15	15
35	15	10	7	62	40	11	8	180	150	15	15
30	16	7	7	55	42	6.5	8	190	160	15	15
30	18	6	7	62	42	10	8	200	170	15	15
35	18	8.5	7	62	45	8.5	8	210	180	15	15
35	20	7.5	7	65	45	10	8	220	190	15	15
40	20	10	7	68	50	9	8	230	200	15	15
35	22	6.5	7	72	50	11	8	250	220	15	15
40	22	9	7	72	55	8.5	8	270	240	15	15
47	22	12.5	7	80	55	12.5	8	300	260	20	20
40	25	7.5	7	80	60	10	8	320	280	20	20
47	25	11	7	85	60	12.5	8	340	300	20	20
52	25	13.5	7	85	65	10	10	360	320	20	20
40	28	6	7	90	65	12.5	10	380	340	20	20
47	28	9.5	7	90	70	10	10	400	360	20	20
52	28	12	7	95	70	12.5	10	420	380	20	20
								440	400	20	20

D_B, Bore diameter; D_S, shaft diameter; R_{CS}, radial cross section; W_s, seal width; W_B, bore width.
$W_B = W_s + 0.9$ mm ($W_s \leq 10$), $W_B = W_s + 1.2$ mm ($W_s > 10$) for recessed bore
Source: Chicago Rawhide.

Table 5.3 Shaft Diameter (D_s) Tolerances

Dimensions in mm		Dimensions in inches	
Shaft diameter D_s	Tolerance	Shaft diameter D_s	Tolerance
$D_s \leq 100$	>±;0.08	$D_s \leq 4.000$	>±0.003
$100 < D_s \leq 150$	±0.10	$4.000 < D_s \leq 6.000$	±0.004
$150 < D_s \leq 250$	±0.13	$6.000 < D_s \leq 10.000$	±0.005
$D_s > 10.000$	±0.010		

Source: Rubber Manufacturers Association.

the seal is replaced. The preparation of the surface of the wear sleeve and the sleeve tolerances should be identical to that of a solid shaft to ensure good sealing. Tolerances for the shaft-wear sleeve diameter (D_s) appear in Table 5.3.

The edge of the shaft must be machined smooth with no nicks or burrs to provide a lead-in chamfer that will protect the seal lip during installation. The chamfer should be machined at an angle of 15--20° with a length of 1.8--3.0 mm (0.070--0.120 in.) (Fig. 5.1). A smooth radius of 3.00 mm (0.125 in.) can be used as an alternative method of providing a lead-in chamfer.

5.2.2 Shaft Finish Specification

The shaft finish of the shaft surface in the area where the lip contacts the shaft is critical to the proper functioning of the sealing system. The shaft must be rough enough to allow the seal lip to bed into the shaft during the early hours of use. The shaft will eventually become smooth, and the seal wear track will become rough. If the shaft surface is too smooth, this bedding-in process will not occur. If the shaft surface is too rough, excessive seal lip wear will occur. Early leakage will result in both cases. The shaft surface must be prepared properly to ensure that machine lead does not exist. Rough machining with a lathe will leave spiral grooves on the shaft, which can screw lubrication from the sump to the air side, causing leakage. When the shaft is reversed, the screw thread will pump inward, preventing leakage; however, contaminants may be screwed under the lip and cause excessive wear of the seal lip and the shaft. These screw threads or spiral grooves are known as machine lead. A plunge grinding operation is recommended to ensure that machine lead is eliminated. Mixed number rpm ratios between the rotating shaft and the grinding wheel are recommended to ensure that a nick on the grinding wheel will not track spirals on the shaft. Grinding should continue until sparkout is observed.

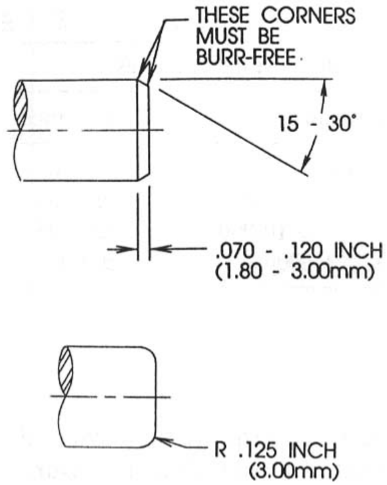


Fig. 5.1 Shaft lead in chamfer. Surfaces and corners must be free of burrs and nicks. (Courtesy of Chicago Rawhide.)

Other methods of shaft finishing, such as peen blasting, paper polishing [130], and honing are often used, but these methods do not ensure the elimination of shaft lead. Transverse grinding should never be used to finish a shaft area where the seal lip will ride because this process will generate shaft lead. Enough grind stock must be left on the shaft after the final machining operation to ensure that the grinding will be deep enough to remove the machining marks.

The proper specification for shaft surface finish is

Roughness 0.25--0.50 μm (10--20 >μin.) R_a (roughness average)
Lead angle 0 ± 0.05°

The angle that the spiral grooves make with the shaft centerline is defined to be the lead angle. Measurement of the shaft surface finish and lead angle is described in Chapter 13 and in Refs. 54, 98, and 150.

5.3 BORE RECOMMENDATIONS AND SPECIFICATIONS

5.3.1 Bore Material and Geometry

The housing bore is the final third of the sealing system [98]. The seal is pressed into a recess in the bore, and the proper fit must be maintained to ensure that no leakage will occur, the seal will not fall out or rotate in the bore, and the seal will not be crushed during installation. Ferrous materials are typically used to form machinery housings, but aluminum is also a popular choice. If the housing materials have different coefficients of thermal expansion, then bore leakage may occur [119]. The recommended tolerances for the bore diameter (D_b) of housings made with ferrous materials are given in Table 5.4. If the housing material is aluminum or some other nonferrous material, then the seal supplier should be consulted.

The edge of the bore must be free of burrs and nicks to prevent damage of the seal outer surface during installation. A chamfer must be provided to help guide the seal into the bore during installation. The length of the chamfer should be 1.5--3.0 mm (0.060--0.090 in.) and should be machined at an angle of 15--30° (Fig. 5.2). If the bore is recessed, then a radius of 0.8 mm (0.030 in.) should be provided at the back of the bore. The bore depth (W_b) depends upon the seal width (W_s) and can be determined with Equations (5.1), (5.2), or (5.3).

5.3.2 Bore Finish Specification

If the seal and bore tolerances are within the recommended specifications and no bore or seal outer diameter (O.D.) damage exists, then no bore leakage will result if the bore surface has a roughness of 2.54 >µm (100 >µin.) R_a or less. If the bore is rougher than 2.54 >µm (100 >µin.) R_a, then a rubber O.D. seal or an O.D. sealant is recommended. The bore roughness should never exceed 3.15 >µm (125 >µin.).

5.4 SEAL SPECIFICATIONS

5.4.1 Dimensional Tolerances

The primary lip-to-shaft interference (I) is calculated with Equation (5.4), where D_L is the primary lip diameter and D_s is the shaft diameter.

The nominal interference may vary considerably from one application to another. A pressure seal with a slowly rotating shaft may have a large interference, and a seal used with a rapidly rotating shaft may have low interference to minimize drag and frictional heat buildup. The control of the primary lip diameter is important to proper seal function; the tolerances are given in Table 5.5. These tolerances are often applied to the secondary lip as well.

Table 5.4 Bore Diameter (D_b) Tolerances for Ferrous Materials

Dimensions in mm		Dimensions in inches	
Bore diameter D_b	Tolerance	Bore diameter D_b	Tolerance
$D_b \leq 50$	+0.039; --0.0	$D_b \leq 3.000$	±0.001
50 $D_b \leq 80$	+0.046; --0.0	3.000 Db ≤7.000	±0.0015
80 $D_b \leq 120$	+0.054; --0.0	7.000 $D_b \leq 12.000$	±0.002
120 $D_b \leq 180$	+0.063; --0.0	12.000 $D_b \leq 20.000$	±0.003:
80 $D_b \leq 300$	+0.075; --0.0	20.000 $D_b \leq 40.000$	±0.004
300 $D_b \leq 440$	+0.084; --0.0	$D_b > 40.000$	±0.006

Source: Rubber Manufacturers Association.

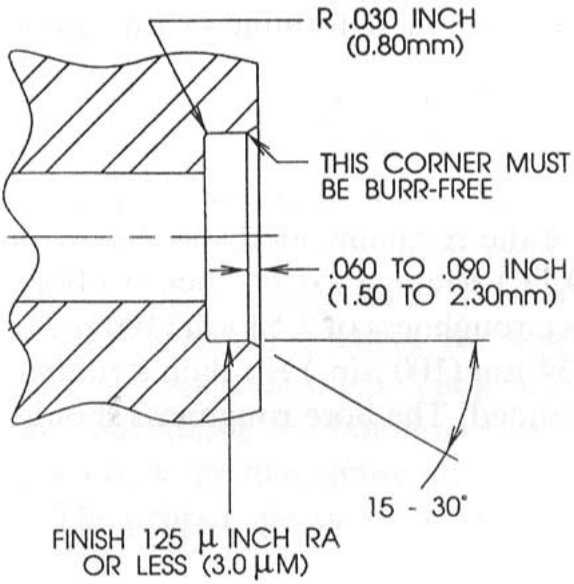

R .030 INCH
(0.80mm)
THIS CORNER MUST
BE BURR-FREE
.060 TO .090 INCH
(1.50 TO 2.30mm)
15 - 30°
FINISH 125 µ INCH RA
OR LESS (3.0 µM)

Fig. 5.2 Bore dimensions. Corners must be free of nicks and burrs. (Courtesy of Chicago Rawhide.)

The radial wall dimension (RWD) is defined by Equation (5.5), where Do is the seal outer diameter and DL is the primary lip diameter.

$$\text{RWD} = \frac{D_o - D_L}{2} \tag{5.5}$$

The radial wall variation (RWV) is the maximum difference between the values of RWD as it is measured around the circumference of the seal. RWV is a measure of how far the center of the circle defining the bore is offset from the center of the circle defining the primary lip. It also deter-

Table 5.5 Primary Tolerances for Lip Inner Diameter

Dimensions in mm		Dimension in inches	
Shaft diameter D_s	Primary lip I.D. tolerance	Shaft diameter D_s	Primary lip I.D. tolerance
$D_s \le 75$	±0.50	$D_s \le 3.000$	±0.020
$75D_s \le 150$	±0.65	$3.000D_s \le 6.000$	±0.025
$150D_s \le 250$	±0.75	$6.000D_s \le 10.000$	±0.030

Source: Rubber Manufacturers Association.

Table 5.6 Maximum Values for Radial Wall Variation (RWV)

Dimensions in mm		Dimensions in inches	
Shaft diameter	Max. RWV	Shaft diameter	Max. RWV
$D_s \le 75$	0.6	$D_s \le 3$	0.025
$75\ D_s \le 150$	0.8	$3\ D_s \le 6$	0.030
$150\ D_s \le 250$	1.0	$6\ D_s \le 10$	0.040

Source: Rubber Manufacturers Association.

Table 5.7 Seal Width (W_s) Tolerances

Dimensions in mm		Dimensions in inches	
Seal width	Tolerance	Seal width	Tolerance
$W_s\ 10$	±0.3	$W_s\ 0.400$	±0.015
$W_s > 10$	±0.4	$W_s > 0.400$	±0.020

Source: Rubber Manufacturers Association.

mines if there is any lobbing of the bore relative to the primary lip. The maximum allowable values for RMV appear in Table 5.6.

The seal width must be controlled to ensure a proper fit into the housing bore. Tolerances for the seal width appear in Table 5.7 [195].

The seal outer diameter must be carefully controlled because the pressfit that results when the seal is installed into the housing bore must prevent leakage. The nominal pressfit (P_f) between the seal outer diameter and the bore inner diameter is defined by the equation

$$P_f = D_o - D_b \tag{5.6}$$

where D_o is the seal outer diameter and D_b is the bore diameter. The seal outer diameter may not be perfectly round. If an excessive out-of-round (OOR) condition exists, then bore leakage may result.

The seal O.D. tolerance, maximum allowable seal out of round, and nominal pressfit are given in Table 5.8 for metal O.D. seals and in Table 5.9 for rubber O.D. seals. It should be noted that the nominal pressfits for rubber O.D. seals are larger than for metal O.D. seals. This allows the rubber at the O.D. to be compressed and generate an adequate sealing force between the housing bore and the sealing O.D.

5.4.2 Radial Load and Lip Opening Pressure Specifications

Seal performance depends upon the sealing force or radial load that the seal lip exerts on the shaft after installation. The force can be estimated by measuring the lip opening pressure (LOP). The LOP is defined as the amount of pressure applied to the air side of the seal that will lift the lip and allow 10,000 cm3/min of air to pass under the lip. This measurement is widely used in industry as a quality check on production parts. It is loosely related to radial load, but a direct correlation from one seal design to another is not possible since the LOP is dependent upon seal geometry as well as on the radial load. The LOP tolerance is ±30% of the nominal pressure reading. The minimum tolerance is ±0.125 bar (±2 psi) [195]. The radial force can be measured directly with a split-shaft device. The tolerances for radial load are given in Table 5.10.

5.5 GARTER SPRINGS

5.5.1 Garter Spring Dimensions and Materials

Garter springs are assembled on radial lip seals to augment the sealing force between the elastomeric lip and the shaft. The spring also compensates for the changes that occur when the elastomer is exposed to lubricants at elevated temperatures. Without the spring, the lip may swell outward and lose interference, and leakage will result. The spring increases the radial stiffness, which increases the ability of the seal to follow shaft dynamic runout [154, 195]. The garter spring is formed by joining a coiled spring into a circle and screwing the nib at one end of the spring into the other end (Fig. 5.3). The maximum allowable gap at the nib is two wire diameters. The ends of the spring should be backwound slightly before assembly to prevent the spring from snapping into a figure eight configuration after assembly. Springs that do form a figure eight must snap back into the original circular shape when dropped approximately 30 mm (12 in.) onto a flat, hard surface. The wire diameter is specified as a reference

Table 5.8 Metal O.D. Seal--O.D. Specifications

Dimensions in mm

Seal O.D. (D_o)	Seal O.D. tolerance	Nominal pressfit	Maximum out of round (OOR)
D_o ≤50	+0.20 +0.08	0.12	0.18
50 D_o ≤80	+0.23 +0.09	0.14	0.25
80 D_o ≤120	+0.25 +0.10	0.15	0.30
120 D_o ≤180	+0.28 +0.12	0.17	0.40
180 D_o ≤300	+0.35 +0.15	0.21	0.25% of Do
300 D_o ≤440	+0.45 +0.20	0.28	0.25% of Do

Dimensions in inches

Seal O.D. (D_o)	Seal O.D. tolerance	Nominal pressfit	Maximum out of round (OOR)
D_o ≤2	±0.002	0.005	0.007
2 D_o ≤3	±0.0025	0.0055	0.010
3 D_o ≤5	±0.003	0.0065	0.012
5 D_o ≤7	±0.003	0.007	0.016
7 D_o ≤12	±0.0035	0.0085	0.0025
12 D_o ≤20	±0.005	0.012	
20 D_o ≤40	±0.005	0.013	
			inch per inch of seal O.D.
40 D_o 60	±0.006	0.016 0.008 0.008	

Source: Rubber Manufacturers Association.

Table 5.9 Rubber O.D. Seals-- O.D. Specifications

Dimensions in mm				Dimensions in inches			
Seal O.D. (DO)	Seal O.D. tolerance	Nominal pressfit	Maximum out of round (OOR)	Seal O.D. (DO)	Seal O.D. tolerance	Nominal pressfit	Maximum out of round (OOR)
$D_o \leq 50$	+0.30 +0.15	0.20	0.25	$D_o \leq 2$	±0.003	0.008	0.010
$50 < D_o \leq 80$	+0.35 +0.20	0.25	0.35	$2 < D_o \leq 3$	±0.003	0.010	0.014
$80 < D_o \leq 120$	+0.35 +0.20	0.25	0.50	$3 < D_o \leq 5$	±0.003	0.0105	0.020
$120 < D_o \leq 180$	+0.45 +0.25	0.32	0.65	$5 < D_o \leq 7$	±0.004	0.012	0.026
$180 < D_o \leq 300$	+0.50 +0.25	0.34	0.80	$7 < D_o \leq 12$	±0.004	0.0125	0.031
$300 < D_o \leq 440$	+0.55 +0.30	0.38	1.00	$12 < D_o \leq 20$	±0.005	0.015	0.039
				$20 < D_o \leq 40$	±0.006	0.018	0.045
				$40 < D_o \leq 60$	±0.007	0.020	0.050

Source: Rubber Manufacturers Association.

Table 5.10 Tolerances for Radial Load (RL)

Dimensions in mm		Dimensions in inches	
Shaft diameter (D_s)	Tolerance	Shaft diameter (D_s)	Tolerance
DS ≤75	RL ± 45%	DS ≤3	RL ± 45%
75‹DS ≤250	RL ± 40%	3‹DS ≤10	RL ± 40%

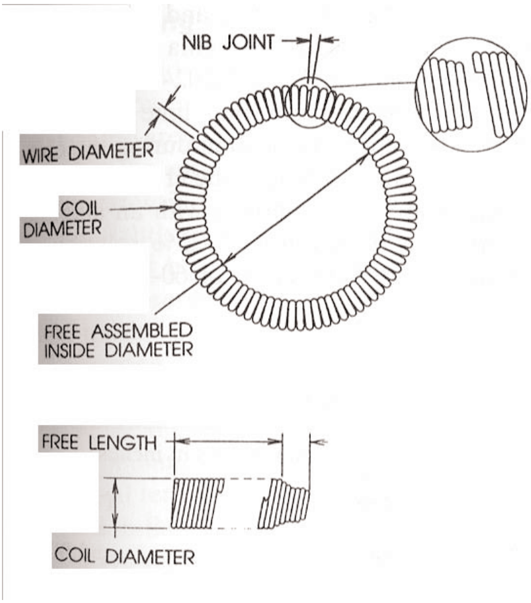

NIB JOINT
WIRE DIAMETER
COIL DIAMETER
FREE ASSEMBLED INSIDE DIAMETER
FREE LENGTH
COIL DIAMETER

Fig. 5.3 Garter spring geometry. Assembled inner diameter (AID) = (free length)/π-- W_d. (Courtesy of Chicago Rawhide.)

dimension to allow the spring manufacturer latitude in achieving the required tension. Tolerances for the wire diameter are listed in Table 5.11. The tolerance for the coil diameter is ±0.13 mm (±0.005 in.). The variation of the coil diameter within a given spring must not exceed 0.08 mm (0.003 in.). The ratio of the coil to wire diameters should exceed 5.3:1 to prevent excessive stresses in the spring.

When the spring is coiled, the edges of the coils on the inner diameter will touch. The diameter that results is called the assembled inner diameter (AID) and can be calculated as

$$\text{AID} = \frac{\text{free length}}{\pi} - W_d \qquad (5.7)$$

The AID is important dimension. When the spring is installed in the spring groove, the spring will be expanded initally and will then pull the seal lip radially inward until the coils touch. The spring AID thus controls the seal free lip diameter. Tolerances for the spring AID are given in Table 5.12. Garter springs are usually made from carbon steel spring wire. Acceptable grades are SAE 1050-1095 (AISI C1050--C1905 and music wire. Stainless steel springs are often used in corrosive media or in high-temperature applications up to 200° C (400° F) (SAE 30302--30304; AISI 302--304). Stainless steel wire is coated with lime and soap, lime and oil, or copper to decrease coiling die wear and supply lubrication during the coiling process. If the stainless steel is passivated to reduce the chemical activity of the surface, then the coatings should be removed with an alkaline cleaner before passivation. The spring is placed in a 15-25% solution of nitric acid in water for about 5 min at a temperature of 60--70° C (140-

Table 5.11 Garter Spring Wire Diameter Tolerance

Dimensions in mm		Dimensions in inches	
Wire diameter (W_d)	Tolerance	Wire diameter (W_d)	Tolerance
0.15--0.25	±0.008	0.006--0.010	±0.0003
0.28--0.38	±0.010	0.011--0.015	±0.0004
0.41--0.48	±0.013	0.016--0.019	±0.0005
0.50--0.69	±0.015	0.020--0.027	±0.0006
0.71--0.86	±0.018	0.018--0.034	±0.0007

Source: Rubber Manufacturers Association.

Table 5.12 Garter Spring Assembled Inner Diameter (AID) Tolerances

Dimensions in mm		Dimensions in inches	
Wire diameter (W_d)	AID	Wire diameter (W_d)	AID
0.15--0.28	±0.20	0.006--0.011	±0.008
0.30--0.48	±0.30	0.012--0.019	±0.012
0.50--0.76	±0.40	0.020--0.030	±0.015
0.80--1.40	±0.50	0.031--0.055	±0.020

Source: Rubber Manufacturers Association.

160° F) to reduce the chemically active surface to a more passive state. A rinse in water should follow the treatment.

Uncoiled springs are often heat-treated to relieve stresses and prevent load loss during high-temperature [100° C (212° F) or more] field operations. The heat-treating temperature is normally 200° C (400° F) for mild steel and 260° C (500° F) for stainless steel. Heat treating will reduce the maximum obtainable initial tension.

5.5.2 Spring Load

A typical load versus deflection chart appears in Fig. 5.4. The curve is generated by cutting the garter spring at a point directly opposite the spring joint or nib area. The ends of the spring are placed in a machine that slowly stretches the spring. The force and deflection are measured and recorded during extension. During the coiling operation, the wire loops are back-wound slightly to provide the initial tension shown in the chart. This initial tension must be exceeded before deflection occurs. The load increase will be directly proportional to the deflection after the applied load exceeds the initial tension and begins to separate the coils.

The spring design should incorporate the smallest wire diameter possible without exceeding the allowable stress limits for the material. This will provide the highest initial tension and the lowest spring rate. A low spring rate will minimize load variation due to spring length and AID variation. A high initial tension also ensures that the coils will touch when the spring is installed in the seal groove and that the seal lip is in the free state before installation on the shaft. The initial tension should be 50--80% of the total load at the working spring deflection when the seal is installed on the shaft.

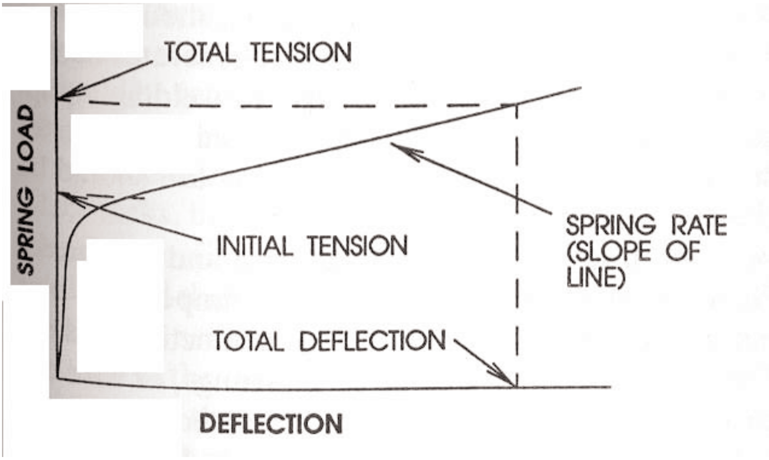


Fig. 5.4 Garter spring load versus deflection. (Courtesy of Chicago Rawhide.)

The tolerance for total load is the greater of ±0.14 N (±0.5 oz) or ±20% of the nominal load at the desired working deflection.

5.6 RECOMMENDED INSTALLATION PROCEDURES

5.6.1 Storage and Handling

Before installation, seals must be packaged, purchased, shipped, inspected, and stored. The type of packaging should be carefully considered by the user and supplier to provide the required protection at the lowest cost [156]. Seals can be bulk packed by randomly jumbling them into a cardboard container. The parts can be stacked, oriented, separated by layers of cardboard, or placed in egg crate dividers. Seals can be roll wrapped in waxed or oiled paper to reduce risk of corrosion. Plastic shrink-wrap material is also popular. The most expensive method of packaging is to individually wrap the seals to prevent corrosion and pack each one in its own box. Some users and suppliers have developed reusable containers in an effort to reduce costs.

Seal boxes are often placed on pallets and covered with a plastic material to resist corrosion. These pallets must be well protected to prevent damage during shipping and warehouse handling. Cartons may be opened and random samples selected for quality control checks (see Chapter 13). The cartons must be opened with care to prevent damage to the rubber parts. The carton should be properly sealed after the samples have been taken to ensure that the remaining parts are not damaged.

Seals can be stored indefinitely if the proper precautions are taken. The storage area should be cool (15--32° C; 60--90° F) and dry, with a humidity of 40--70%. To prevent premature aging of the material, seals should not be stored near heat. Some seal materials such as nitrile (NBR) can be adversely affected by ozone. Open boxes of these materials should not be stored in direct sunlight or near electrical equipment that generates ozone. Even short exposure of unprotected seals stored close to welding equipment can create ozone cracks in nitrile elastomers. To minimize risks of deterioration during storage, the first seals into the warehouse should be the first ones used in the manufacturing area.

After the seals have been drawn out of the warehouse and unpacked, they must be handled carefully before installation. Seals dumped into bins may become permanently damaged by the metal cases contacting and deforming adjacent seal lips. Hanging seals on pegs, nails, strings, or wire can also cause distortion or even cut the lip. Seals should be handled by their outer surface to prevent damage to the sealing lip. Lip seals dumped onto surfaces such as workbenches are subject to contamination. Seals that have been prelubed are particularly susceptible to this hazard. These surfaces should be free of grit, chips, and other abrasives.

If seals do get dirty, they must be cleaned carefully. Abrasive cleaners should never be used. Some cleaners may cause the elastomers to swell excessively, harden, or disintegrate. The elastomer may also be debonded from the case, rubber O.D. sealants may wash away, and metallic parts such as the spring or case may be damaged. Environmentally safe cleaners should always be used. The seal supplier should be consulted to determine what cleaners are recommended for a given seal material.

5.6.2 Preinstallation Practices

Seal installation in the field environment is a much different process than installation in the factory [98]. The factory environment is typically cleaner than the field environment, and the installation tools are well designed and maintained. Hydraulic presses are used to push the seals squarely into the housing with a controlled force to prevent crushing of the seal case. The process is often automated. A factory-installed seal will rarely leak prematurely because of improper installation.

Improper field installation is one of the main causes of premature seal failure. The first step to guarantee a good installation is to check the seal dimensions to be sure the seal will fit the application. If possible, the replacement seal part number should match the part number of the original equipment seal. Sometimes the seal dimensions may be identical, but the material or lip design may not meet the requirements of the application. The seal should then be examined for damage prior to installation. A seal lip will leak if is turned back or nicked. The seal outside diameter should be checked for dents, scores, or cuts. Replace a damaged seal. Never reuse a seal; always install a new one. Be sure the primary seal lip faces the same direction as the old one. Seals that are installed backward will leak instantly. The edge of the housing must be inspected, and any burrs that may damage the seal O.D. should be removed. The bore edge should be rounded or chamfered and the inside of the bore finished to the proper specifications.

The shaft should be examined and repaired or replaced if required. Surface nicks, burrs, and grooves must be removed without generating machine lead. If possible, the shaft should be checked to ensure that no machine lead exists and that the surface finish is 0.25--0.50 μm (10--20 μin.) Ra in the area where the seal lip will contact the shaft. If the shaft is grooved, a thin spacer is placed in the back of the bore so the new seal lip will ride on an undamaged portion of the shaft. Sometimes the equipment manufacturer will install a wear sleeve over the shaft to provide a running surface for the seal. This wear sleeve must be replaced before the seal is installed. The old one must be carefully removed by heating the wear ring and gently tapping it off the shaft. In some cases, the wear sleeve must be chiseled off. If the shaft is damaged, it is recommended that the nicks be filled with epoxy then sanded down smooth before the wear sleeve is installed. The wear sleeve is often heated before it is slipped over the shaft. The heat causes the wear ring to expand, which makes it easier to install over the cool shaft. As the ring cools, it contracts around the shaft and minimizes leakage between the wear ring and the shaft. Sometimes a sealant is placed between the shaft and the wear ring. Care must be taken to ensure that the material does not smear on the surface of the wear ring. If the original manufacturer did not provide a wear ring, then the seal user should not attempt to install a seal with the original part number over an improvised thick-walled wear sleeve. The seal lip will distort, and premature leakage will result. Thin-walled wear sleeves (0.25 mm; 0.010 in.) can be used with a new seal that has the original part number without significant reduction in seal life. Grooves in the shaft should be filled with epoxy and sanded smooth before the thin-walled wear sleeve is installed. This will prevent the wear sleeve from distorting into the groove. The seal lip should be lubricated with the oil or grease from the application to assist in installation.

5.6.3 Installation Tools

Installation tools are used to prevent lip damage and ensure that the seal is bottomed in the bore without cocking. The tools are typically steel and are well designed for factory use. Tools are often lacking or inadequate when used to retrofit seals in the field. The outside diameter of tools used to press the seal in the bore should be about 0.25 mm (0.010 in.) smaller than the housing bore. The tool should always press near the seal O.D. because this is usually the strongest region and distortion will be minimized. If the tool presses near the seal I.D., case distortion may easily occur and early leakage will result.

If the housing has a step, then the tool should be designed to bottom the seal in the step to prevent cocking (Fig. 5.5a). The back side of the step should be machined square to the housing bore. If the housing has no step, then the tool must be designed to bottom against the shaft or housing face front. If the tool bottoms against the shaft face front, then this surface must be machined perpendicular to the shaft centerline to minimize seal cocking (Fig. 5.5b). When the tool bottoms against the housing face, then this face must be machined square with the bore (Fig. 5.5c). In the factory, hydraulic or pneumatic presses are used with the proper installation tools to install seals. Pressure is carefully regulated to prevent damage. In the field, hand-operated arbor presses (Fig. 5.6) or a soft-faced mallet (Fig. 5.7) are typically used to press the seal into the bore.

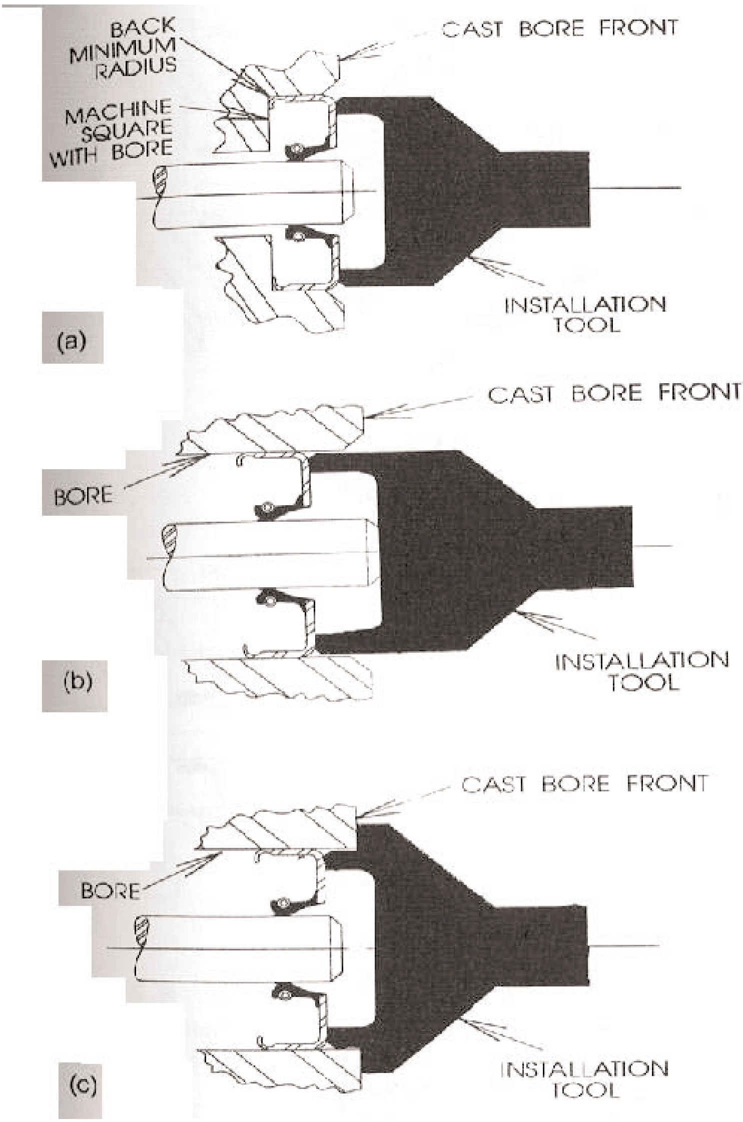

BACK
MINIMUM
RADIUS
MACHINE
SQUARE
WITH BORE
CAST BORE FRONT
INSTALLATION
TOOL
(a)
CAST BORE FRONT
BORE
INSTALLATION
TOOL
(b)
CAST BORE FRONT
BORE
INSTALLATION
TOOL
(c)

Fig. 5.5 Installation tools. (a) Stepped bore, where the back of the bore is machined with the bore. (b) Through bore tool bottoms on shaft face-- shaft face is machined perpendicular to shaft centerline. (c) Through bore tool bottoms on housing face-- bore face machined square with bore.

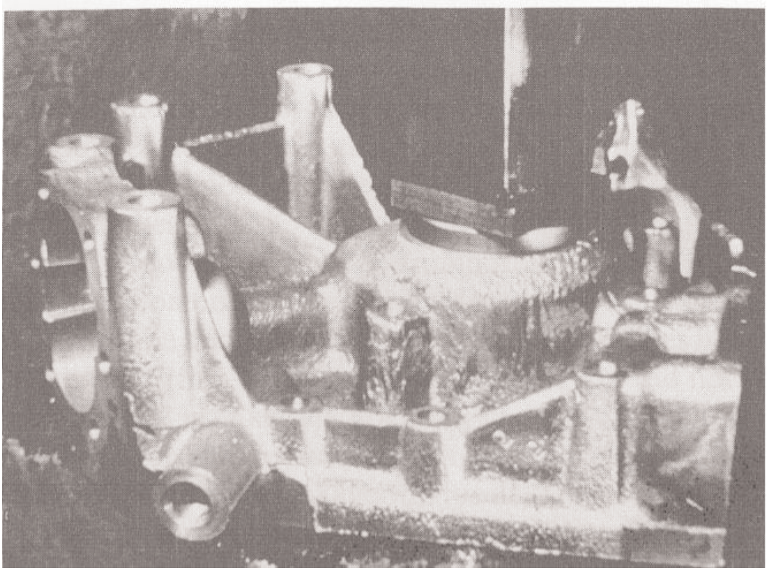


Fig. 5.6 Arbor press used for field installation.

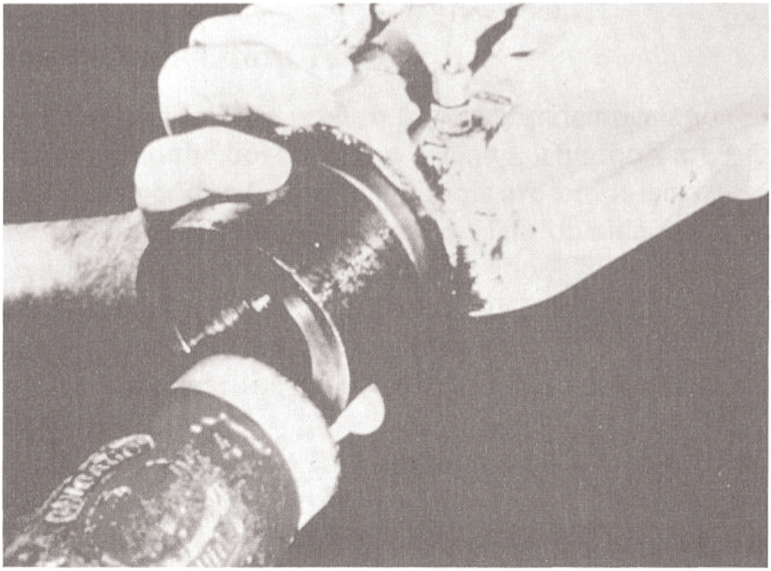


Fig. 5.7 Soft-faced mallet used for field installation.

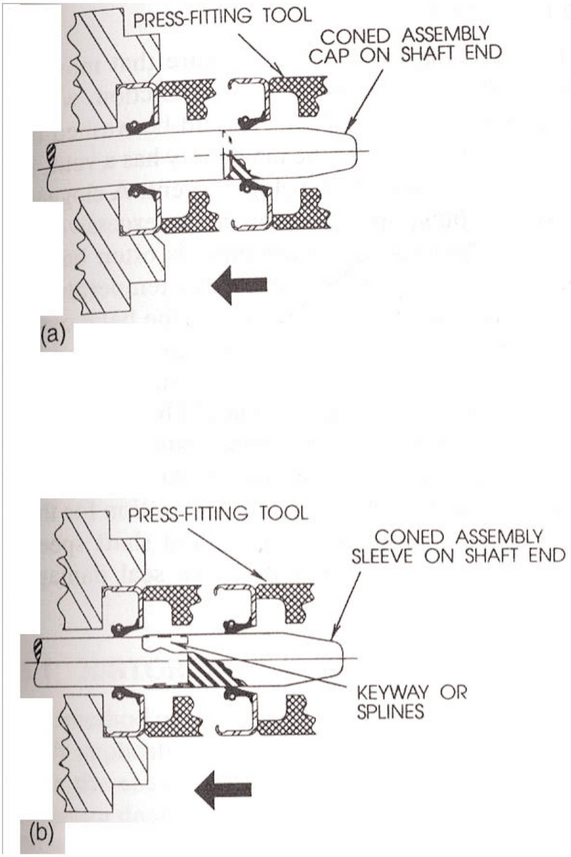

PRESS-FITTING TOOL
CONED ASSEMBLY CAP ON SHAFT END
(a)
PRESS-FITTING TOOL
CONED ASSEMBLY SLEEVE ON SHAFT END
KEYWAY OR SPLINES
(b)

Fig. 5.8 Assembly cones. (a) Cone at shaft end protects seal against burrs and nicks at shaft end, provides chamfer if one is missing. (b) Cone covers splines and keyways. (Courtesy of Chicago Rawhide.)

Shafts often have keyways or splines, and assembly cones are used with the installation tools to prevent lip damage. The cone can be designed to fit over the edge of the shaft (Fig. 5.8a) or extend completely over the spline and keyway regions (Fig. 5.8b). During installation, the lip may tend to invert, which could dislodge the spring or turn the lip over completely. Instant leakage would result. The seal lip and/or cone should be lubricated to prevent inversion, The cones are sometimes made from low-friction materials (PTFE, nylon) and help prevent lip inversion.

5.6.4 Postinstallation Practices

After the installation has been completed, check to be sure that moving machinery parts will not rub on the seal, creating excessive friction, wear, and heat. If the machinery is painted, the seal area should be washed to prevent paint from covering the seal lip area. If the machinery has a vent, it should also be masked to be sure the paint will not clog the vent. A clogged vent will allow internal pressure to build up, which can create excessive lip wear or blow the seal out of the housing. The machinery is often baked after painting. The seal should not be exposed to excessive temperatures for long periods of time. Machinery should never be left in the bake oven any longer than necessary as severe material degradation could occur. If the area around the seal is cleaned, then cleaning fluids that are not harmful to the elastomers should be used or the seal area should be masked to prevent exposure. After assembly, machines are often tested. In many cases, oil is not present in the seal area initially and it may take 5--10 minutes running time before oil splash provides enough lubrication for the seal. The seal should be lubricated before installation, and shaft speed should gradually build up to full speed to prevent severe seal damage during the initial run-in.

6

Principles of Seal Operation

6.1 HISTORICAL OVERVIEW

The method of preventing leakage with an elastomeric lip seal in a rotating shaft application appears to be obvious. The lip and spring are stretched over the shaft, and a force is generated between the lip and shaft that creates a dam to prevent leakage. The flexibility of the lip-and-spring combination will maintain the dam even if the shaft has vibrations and/or dynamic runout. When the mechanism is closely examined, several questions arise. Is the lip lubricated by a thin film of oil that separates the lip contact region from the shaft, and if so, what prevents the oil from leaking out?

Traditional hydrodynamics and bearing theory cannot provide answers to these questions, because they predict axial leakage if a lubricating film separates moving parts. Researchers have studied this problem for many years, and the sealing mechanism is not yet completely understood [110, 128]. This lack of understanding results in costly trial-and-error development exercises when new materials and designs are required to solve new application problems. The first step to understand the sealing mechanism was to convince doubters that the lip was lubricated [19, 28, 211]. In 1957, Jagger [101, 104] demonstrated that the seal lip was lubricated by a thin film of oil, which was assumed to be 0.001--0.002 mm (40--80 μin.) thick. Modern researchers are attempting to measure this film thickness [41, 111] and relate it to seal torque and performance. Ishiwata and Hirano [94, 96] estimated the pressure distribution between the seal lip and the shaft and predicted the frictional torque of the seal. They believed that the load between the lip and the shaft was supported by the shaft roughness.

Jagger [105] published results that were contrary to this hypothesis. Trial and error had shown that the best sealing performance occurred when the shaft surface was finished to a roughness of 0.25--0.50 μm (10--20 μin.). This initial roughness breaks through the rubber skin at the sealing tip and creates a wear band on the seal lip. A lubricating film then develops that minimizes further lip wear (Fig. 6.1) after a 20--100-hr break-in period. Jagger observed a rough surface on the seal lip with undulations that he called microasperities. If the shaft was too smooth, the seal wear band would not develop. If the shaft was too rough, then the seal lip would be severely damaged before the protective lubricating film could be developed. He also observed that the shaft surface under the seal lip was polished smooth (Fig. 6.2) and concluded that the microasperities on the seal lip supported the load by acting as micro bearing pads [103].

Johnston [108] refined this concept and tried to relate seal torque and film thickness to the dimensions of the rubber microasperities. Jagger [102, 106], Iny and Cameron [92], Rajakovics [146], and others believed that surface tension prevented the lubricating fluid between the seal lip and the shaft from leaking to the atmosphere. This concept was widely accepted until 1978, when Kawahara and Hirabayshi [64, 114] reported that seals would leak profusely when the lip geometry was reversed (i.e., with the

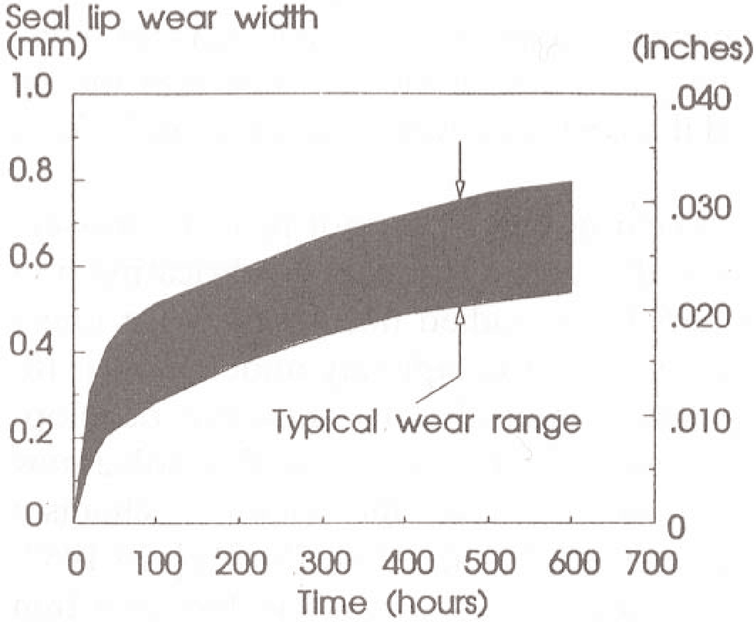


Fig. 6.1 Seal lip wear track versus time. (Courtesy of Chicago Rawhide.)

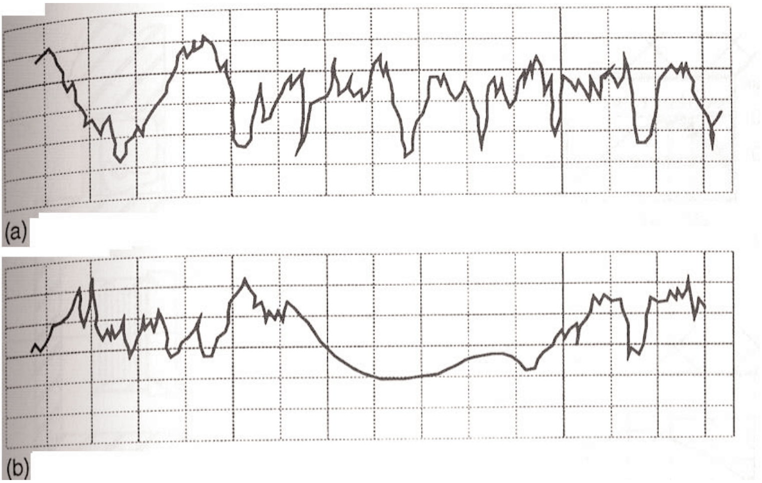


Fig. 6.2 Shaft surface roughness versus time. (a) Before test: 0.25--0.50 μm Ra. (b) After test: seal lip has polished shaft. (Courtesy of Chicago Rawhide.)

spring facing the air side instead of the oil side). They measured the rate of oil flow with the seals conversely installed and found that the leakage depended upon shaft speed, viscosity, surface condition, and running time. The frictional torque of the converse-mounted leaking seal was lower than the frictional torque of the same seal installed in the proper condition without leakage. The tangential displacement of the lip contact region was observed through a glass shaft. A pumping seal has an asymmetric deformation with the largest angle of displacement toward the oil side. A seal that does not pump has a symmetrical displacement (Fig. 6.3). These observations suggest that the contact pressure maximum is closer to the oil side than the air side [97].

Research by Müller and Ott [129, 133] describe another dynamic sealing effect. A wide gap between a rigid sleeve and a rotating shaft will not leak at certain speeds even if the oil sump is flooded. The rotating shaft creates Görtler-Taylor vortices next to the seal contact area that create an inward pumping action that prevents leakage. Further studies by Qu [145] showed that leakage at slow speeds would diminish, stop at a critical speed, and then begin again. It is concluded that vortices are a secondary effect and are not the primary sealing mechanism.

Arai [6] observed that plain lip seals would pump small quantities of oil from the air side to the oil side even if no oil was in the sump. This

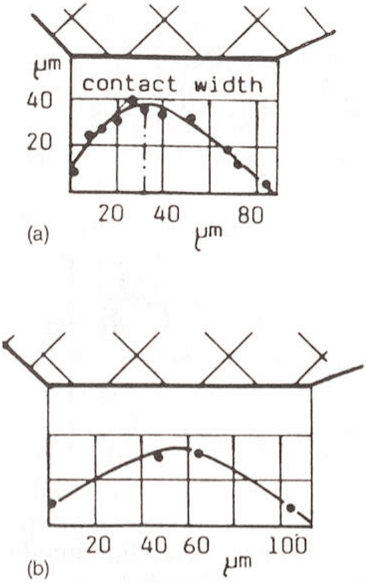


Fig. 6.3 Tangential displacement of seal contact region on a glass shaft. (a) Asymmetric tangential displacement--seal pumps wells. (b) Symmetric tangential displacement--seal does not pump. (Courtesy of BHR.)

eliminated the vortex concept as a primary mechanism in promoting seal pumping. The pump rate increased with increasing shaft speed and decreasing oil viscosity. Pump rate also increased slightly as the dynamic shaft eccentricity increased.

Studies were begun [42, 43, 62, 93, 95, 127] to determine if the pumping phenomenon was dependent on an axial scrubbing action due to the dynamic runout. Horve [88] conducted tests to determine how shaft eccentricity and axial scrubbing affects seal pump rates. A fixture (Fig. 6.4) was constructed that would create a dynamic runout condition to cause axial scrubbing without rotary motion. Very little transfer of oil from the air to the oil side was observed. It is concluded that dynamic runout has very little effect on seal pump rates and that excessive dynamic runout combined with speed can lead to instant failure [82, 83].

Other geometrical conditions can affect seal pump rate. Fazekas and Kalsi [37] described an O-ring that is installed in a groove that is slanted or cocked with respect to the shaft axis. An inward pump rate is generated that reduces leakage in high-pressure rotary shaft applications. Hermann and Seffler [52] reported that a shaft that is tilted or cocked with respect to

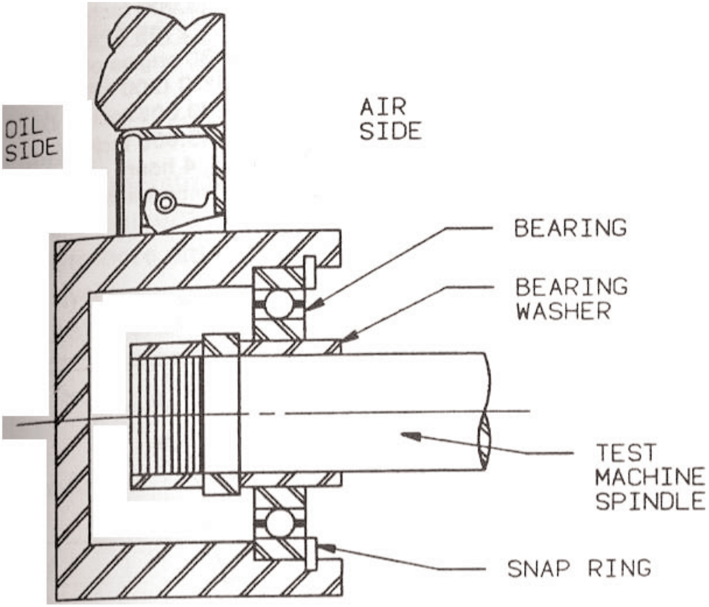


Fig. 6.4 Test fixture to simulate dynamic runout without shaft rotation. (Courtesy of Chicago Rawhide.)

the seal contact band increases the inward pumping ability. The net effect of these phenomena is to generate a one-cycle sinusoidal contact path on the shaft. Horve and Brink [70] introduced a seal with a molded wavy sealing edge that pumps more than conventional plain lip designs.

The conclusion that can be drawn from the years of research is that the radial lip seal has a lubricated lip and that when it is properly designed with a properly compounded material a microscopic pump rate is developed that forces minor leakage back into the sump. In many cases, the seal will ingest air as well [38]. Macroscopic or geometric effects such as helices augment the microscopic pumping ability and increase reliability [227].

6.2 MICROSCOPIC SEALING MECHANISM

Many researchers have reported the presence of microasperities in the wear track of reliable, functioning radial lip seals that exhibit good performance [9, 65, 113, 115, 128, 207]. Horve [89, 90] described the correlation of radial lip seal performance to the lip seal wear condition. Life tests were run with two materials made from the same NBR polymer but with different formulations. The seal design was the same, the shaft size was 76.2 mm (3.000 in.), the lubricant was SAE 30 engine oil from the same drum, and the sump temperature was 93°C (200°F). Nitrile A had excellent perfor-

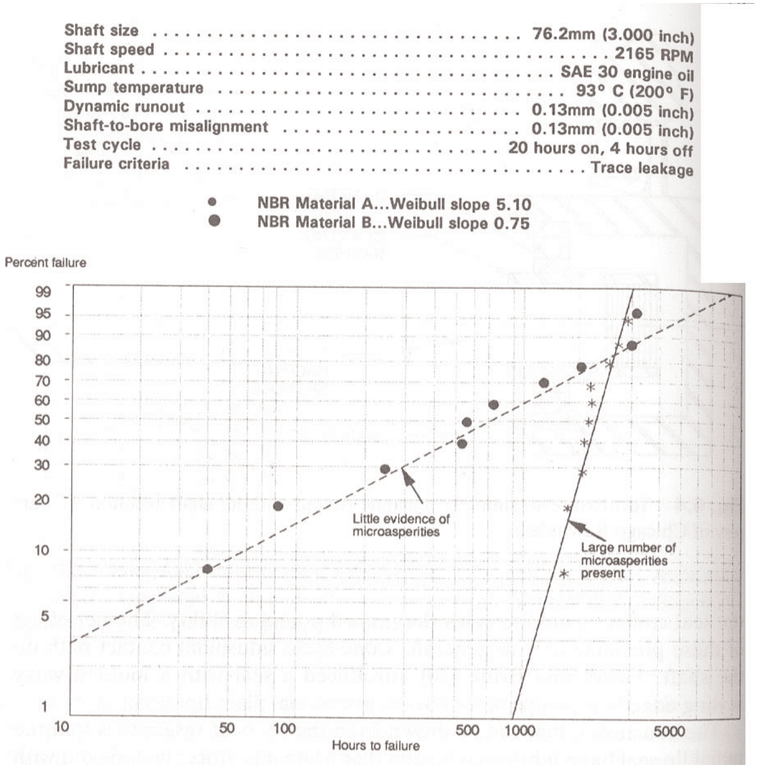


Fig. 6.5 Life characteristics of two nitrile (NBR) compounds. Conditions listed in Table 6.1 (*) NBR A, Weibull slope (●) NBR B, Weibull slope 0.75. (Courtesy of Chicago Rawhide.)

mance with a large Weibull slope. The other material (nitrile B) had poor performance with a low Weibull slope (Fig. 6.5).

All of the nitrile A seals had hardened, lost their interference with the shaft, and leaked upon start-up because they could no longer follow the shaft vibrations and dynamic runout. The seals and material had reached their ultimate life, which is confirmed by the high Weibull slope (5.1) for this group of seals. Most of the seals made with the nitrile B material leaked well before hardening had occurred and a large amount of interference was still remaining (Table 6.1). The wear tracks of the seals were examined with a scanning electron microscope at 300x. The reliable seals made with nitrile A had a rough wear track with an abundance of micro-

Table 6.1 Service Reliability of Two Nitrile (NBR) Compounds

Shaft size	76.2 mm (3.000 in.)
Shaft speed	2165 rpm
Lubricant	SAE 30 engine oil
Sump temperature	93°C (200°F)
Dynamic runout	0.13 mm (0.005 in.)
shaft-to-bore misalignment	0.13 mm (0.005 in.)
Test cycle	20 hr on, 4 hr off
Failure criteria	Trace leakage

NBR A		NBR B	
Time to leak (hr)	Seal lip interference remaining (%)	Time to leak (hr)	Seal lip interference remaining (%)
1569	6.1	40	87.5
1569	7.7	82	77.1
1797	5.8	244	65.2
1877	4.1	534	29.8
1941	3.8	564	53.2
2001	0.0	742	38.3
2021	0.0	1237	20.8
2397	0.0	1810	10.4
2665	4.1	3046	8.7

Source: Chicago Rawhide.

asperities. The unreliable seals made with nitrile B had no microasperities (Fig. 6.6) and exhibited a smooth planar wear track.

Pump tests were performed on both trimmed and molded lip seals made from the two materials. Both trimmed and molded lip seals made with material A had high pump rates with a large number of microasperities present. Material B had poor pump rates without microasperities, and the results were the same for molded and trimmed lip seals (Fig. 6.7).

Molded lip seals made with the poorly pumping nitrile B material pump were tested to establish the pump rate. The seals were removed, and the wear track was artificially roughened with emery cloth. The seal was reinstalled on the same shaft, and the pump tests were repeated. The pump rate increased dramatically but decayed rapidly back to the original value (Fig. 6.8). Examination of the wear track under magnification after test shows that the portion contacting the shaft has begun to glaze over and become smooth (Fig. 6.9). As the wear track becomes smooth, the pump rate diminishes.

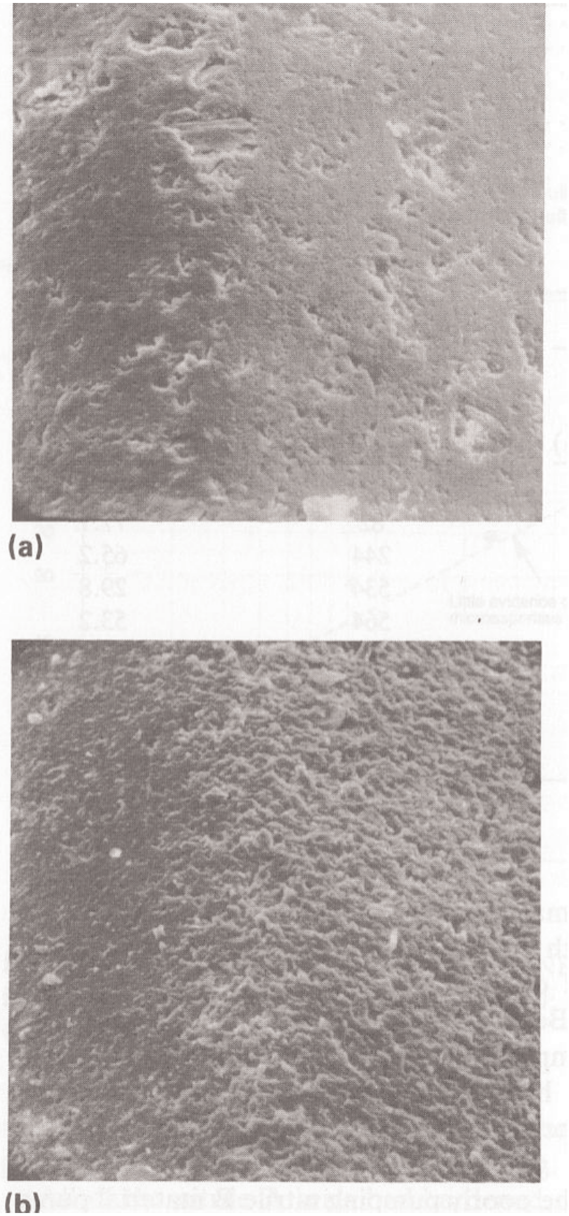


Fig. 6.6 Seal wear track. (a) Nitrile A rough wear track. (b) Nitrile B smooth wear track. Comparison of two different nitrile (NBR) formulations. (300x; 17 hr, shaft surface worn smooth). (Courtesy of Chicago Rawhide.)

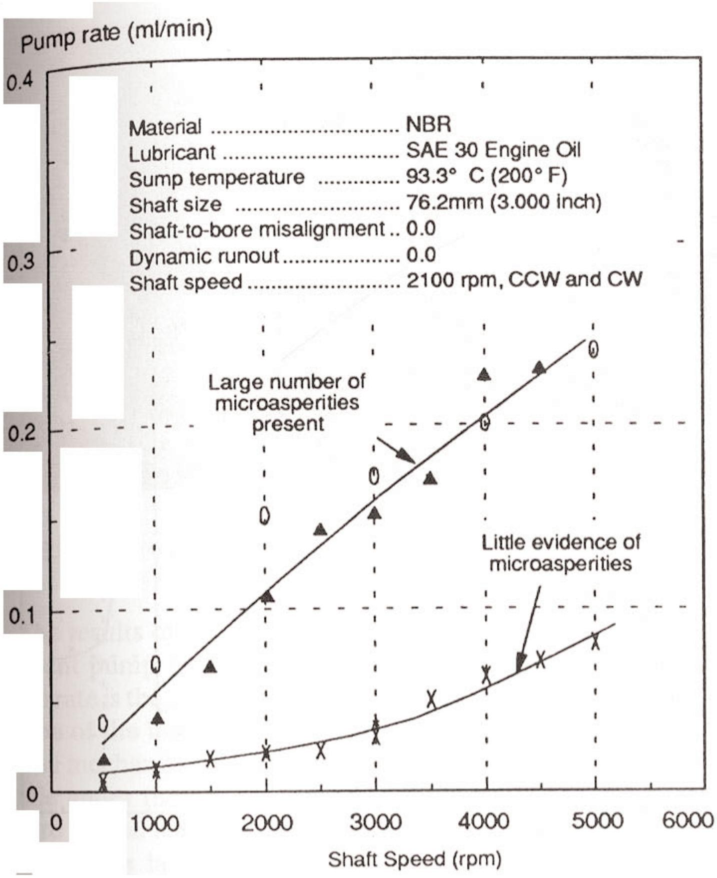


Fig. 6.7 Pump test results of molded and trimmed lip seals made with two different nitrile (NBR) materials. Molded lip: (▲) NBR A; (x) NBR B. Trimmed lip: (o) NBR A; (*) NBR B. (Courtesy of Chicago Rawhide.)

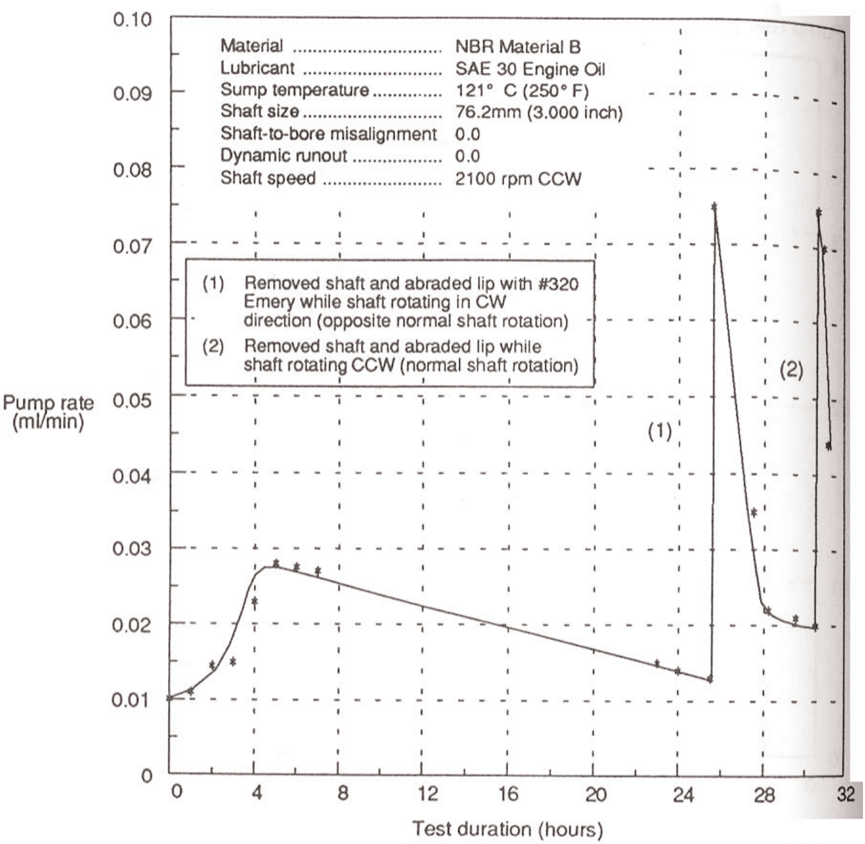


Fig. 6.8 Effect of artificially induced microasperities on seal pump rates. (Courtesy of Chicago Rawhide.)

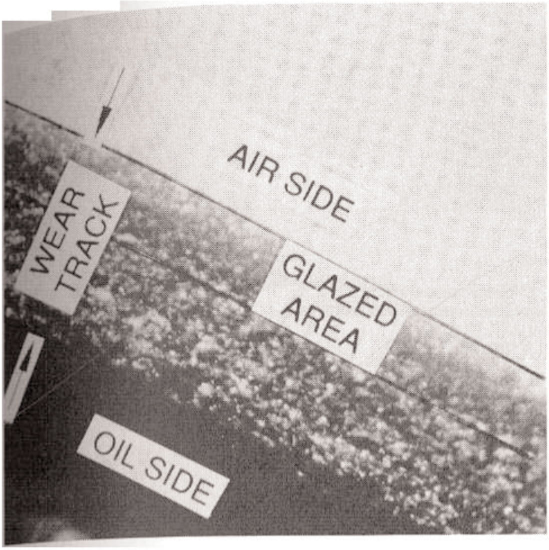


Fig. 6.9 Seal wear track with artificially induced asperities.

The results of the study led to the conclusion that reliable seals have an inherent pump rate that transfers oil from the air side to the sump. This pump rate is the result of microasperities that form in the wear track. The formation of the microasperities is dependent upon the material formulation.

The mechanism can be explained [67, 89, 90, 128]. The contact pressure profile under the seal lip is asymmetric, with the highest pressure gradient toward the oil side. The gradient is highest on the oil side because the oil-side angle is larger than the air-side angle and the spring centerline is positioned to maintain a low air-side angle. The rough shaft removes chunks of material from the rubber surface if the material has been property formulated. This process smoothes the shaft and creates a seal wear track that is full of microasperities. The microasperities provide lubrication pockets to prevent further lip wear. As the shaft rotates, the microasperities are sheared in the tangential direction. A series of miniature helices are formed, and the helix set on the oil side has a larger angle than the helix set on the air side. The result is a net pump rate toward the oil side (Fig. 6.10).

6.3 MACROSCOPIC SEALING MECHANISM

Researchers report that the inherent microscopic seal pump rates are enhanced if the shaft is tilted relative to the seal [52] or if the seal is cocked relative to the shaft [37]. Horve [88] has proposed a theory to explain the macroscopic pumping action. The contact pattern of the misaligned seal

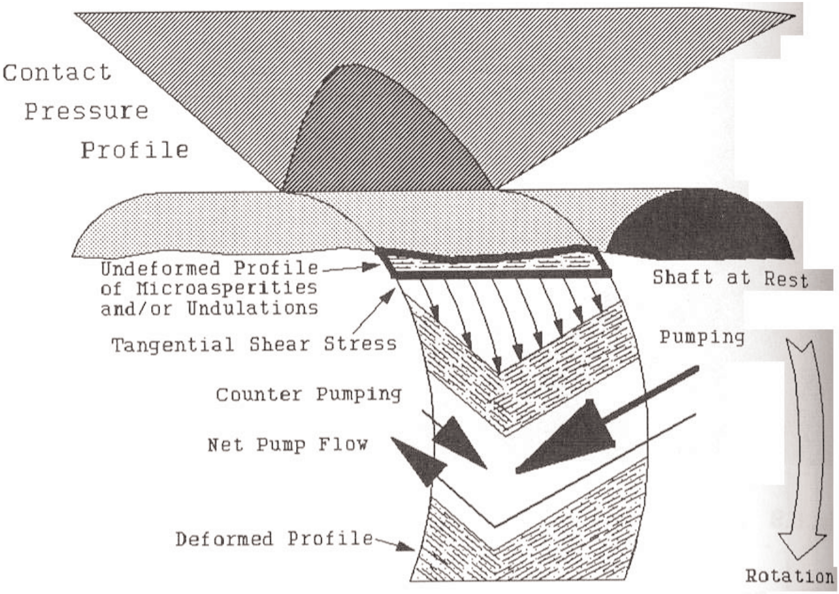

Contact
Pressure
Profile
Undeformed Profile
of Microasperities
and/or Undulations
Tangential Shear Stress
Counter Pumping
Net Pump Flow
Deformed Profile
Shaft at Rest
Pumping
Rotation

Fig. 6.10 Micro visco seal. (Courtesy of BHR.)

forms a single sine wave pattern. Some seal manufacturers design seals with multiple sine waves that contact the shaft (Fig. 6.11) [70]. Increasing the wave amplitude and/or the number of cycles increases the pump rate [69].

The geometry of the seal contact region is illustrated in Fig. 6.12. Linear axial pressure distributions are assumed to exist under the seal lip. Good seal design practice dictates that the oil-side angle exceed the air-side angle. This results in an oil-side pressure gradient that is larger than the air-side pressure gradient. These pressure gradients also depend on radial load seal contact width.

The flow of oil into or out of the sealing area is calculated by using formulas describing flow in a thin circular annulus. The annulus exists because of hydrodynamic lift. The oil film thickness is calculated with Petroff's equation for a journal bearing:

$$h = \frac{4\,\pi^2 Z_{cp} R_s^3 Nw}{Tor} \tag{6.1}$$

The seal torque must be measured and the fluid viscosity at the temperature of the film must be known in order to make this calculation.

The amount of fluid entering the sealing area is determined by the component of shaft velocity perpendicular to the sealing lip. The magnitude of this velocity component depends on the amount of misalignment

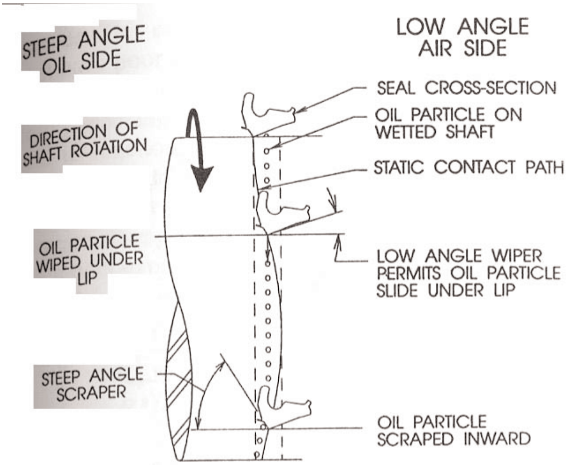

STEEP ANGLE OIL SIDE
LOW ANGLE AIR SIDE
SEAL CROSS-SECTION
OIL PARTICLE ON WETTED SHAFT
DIRECTION OF SHAFT ROTATION
STATIC CONTACT PATH
OIL PARTICLE WIPED UNDER LIP
LOW ANGLE WIPER PERMITS OIL PARTICLE SLIDE UNDER LIP
STEEP ANGLE SCRAPER
OIL PARTICLE SCRAPED INWARD

Fig. 6.11 Seal design with wavy contact on the shaft. (Courtesy of Chicago Rawhide.)

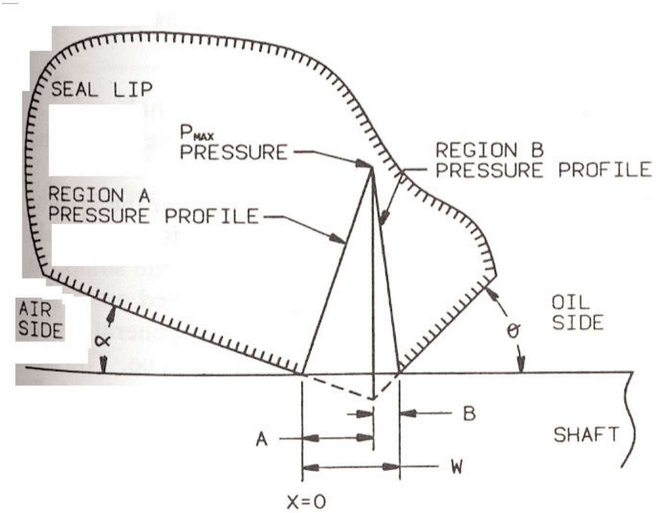

SEAL LIP
P MAX PRESSURE
REGION B PRESSURE PROFILE
REGION A PRESSURE PROFILE
AIR SIDE
α
θ
OIL SIDE
B
A
SHAFT
W
X=0

Fig. 6.12 Pressure distribution under the seal lip. (Courtesy of Chicago Rawhide.)

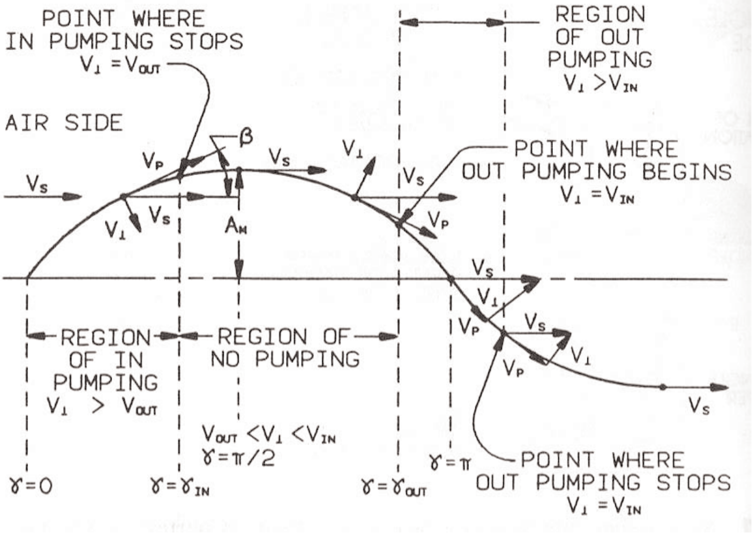

POINT WHERE IN PUMPING STOPS
$V_\perp = V_{OUT}$
AIR SIDE
β
V_P
V_S
$V_\perp$
A_M
REGION OF OUT PUMPING
$V_\perp > V_{IN}$
POINT WHERE OUT PUMPING BEGINS
$V_\perp = V_{IN}$
REGION OF IN PUMPING
$V_\perp > V_{OUT}$
REGION OF NO PUMPING
$V_{OUT} < V_\perp < V_{IN}$
$\gamma = \pi/2$
$\gamma = 0$
$\gamma = \gamma_{IN}$
$\gamma = \gamma_{OUT}$
$\gamma = \pi$
POINT WHERE OUT PUMPING STOPS
$V_\perp = V_{IN}$

Fig. 6.13 Velocity components along wave with amplitude Am and n cycles. V_1, shaft velocity component perpendicular to seal lip; V_p, shaft velocity component parallel to seal lip; V_s, shaft linear velocity. (Courtesy of Chicago Rawhide.)

between the seal circumferential contact pattern and the shaft axis. A cocked seal lip forms a single sine wave contact pattern on the shaft. The amplitude is one-half the magnitude of the cocking.

In one portion of the wave, the component is directed under the lip into the sump (Fig. 6.13). When the pressure generated exceeds the air-side pressure gradient in the contact zone, the oil will flow into the sump. In another portion of the wave, the velocity component is directed under the lip out of the oil sump. Leakage will occur if the pressure generated exceeds the oil-side underlip pressure gradient in the contact zone. The net pump rate is the sum of the differences between the flow due to the shaft velocity components and the flow due to the underlip pressure gradients along the entire seal circumference.

The net pump rate is given by Equations (6.2)-(6.10). As the shaft speed approaches infinity, this equation will reduce to a simpler form that represents the maximum possible macroscopic pump rate (Equation 6.11). The model explains many of the observations by researchers. When the air-side angle (α) is larger than the oil-side angle (θ), the model predicts that the outward flow will exceed the inward flow. This agrees with observations that gross leakage occurs when the seal is installed backward. The model predicts that pump rates will increase as the shaft seal misalignment increases. This explains the erratic behavior sometimes observed with seals that have poor inherent pumping abilities and a lack of microasperities in the seal wear track (Table 6.1; Fig. 6.5). An imperfect seal installation can lead to cocking, which generates a macroscopic pump rate that allows the seal to function longer than one with a non-cocked installation. Model predictions agree with the trends discussed in Chapter 8. Pump rates increase as shaft speed, shaft size, sump temperature, and seal misalignment increase. Pump rates also increase as the fluid viscosity decreases. Finally, the model explains why seals with molded waves have higher pump rates than plain lip seals.

$$V_{\text{in}} = \frac{R_L h^2}{6Z_{\text{cp}} w^2} \left(\frac{\tan\alpha + \tan\theta}{\tan\alpha} \right) \tag{6.2}$$

$$V_{\text{out}} = \frac{R_L h^2}{6Z_{\text{cp}} w^2} \left(\frac{\tan\alpha + \tan\theta}{\tan\theta} \right) \tag{6.3}$$

$$C = \frac{\text{Am}\, n}{R_s} \tag{6.4}$$

$$V_s = 2\pi R_s N \tag{6.5}$$

$$A_{\text{in}} = \frac{1 - C^2 + \dfrac{2}{[(V_s/V_{\text{in}})^2 - 1]}}{1 + C^2} \tag{6.6}$$

$$A_{\text{out}} = \frac{1 - C^2 + \dfrac{2}{[(V_s/V_{\text{out}})^2 - 1]}}{1 + C^2} \tag{6.7}$$

$$\gamma_{\text{in}} = \arccos\left(\frac{1}{C\left[\left(\dfrac{V_s}{V_{\text{out}}}\right)^2 - 1\right]^{1/2}} \right) \tag{6.8}$$

$$\gamma_{\text{out}} = \arccos\left(\frac{1}{C\left[\left(\dfrac{V_s}{V_{\text{in}}}\right)^2 - 1\right]^{1/2}} \right) \tag{6.9}$$

$$Q_{\text{net}} = 2\pi h R_s V_s \left\{ \frac{(\arcsin(A_{\text{in}}) - \arcsin(A_{\text{out}})}{2\pi} + \left[\frac{V_{\text{in}}}{V_s}\left(1 - \frac{\gamma_{\text{out}}}{\pi}\right) - \frac{V_{\text{out}}}{V_s}\left(\frac{\gamma_{\text{in}}}{\pi}\right) \right] \right\} \tag{6.10}$$

$$Q_{\text{net}\infty} = \frac{\pi h^3 R_s R_L}{6Z_{\text{cp}} w^2} \left(\frac{\tan^2\theta - \tan^2\gamma}{\tan\theta \tan\gamma} \right) \tag{6.11}$$

where

α = air-side angle (degrees)
θ = oil-side angle (degrees)
γ = point along seal contact band
γ_{in} = point along seal contact band where in-pumping stops (degrees)
γ_{out} = point along seal contact band where out-pumping begins (degrees)
A_m = amplitude of wave on seal lip (mm)
A_{in} = dimensionless parameter
A_{out} = dimensionless parameter
C = dimensionless parameter
h = film thickness (mm)
n = number of waves on seal lip
N = shaft speed (rev/sec)
Q_{net} = net volumetric flow into/out of the sump (mm^3/sec)
$Q_{net\infty}$ = net volumetric flow at infinite shaft speed (mm^3/sec)
R_L = radial load (N/mm)
R_s = shaft radius (mm)
Tor = seal torque (N -- mm)
V_{in} = inward velocity (mm/sec)
V_{out} = outward velocity (mm/sec)
V_s = linear shaft velocity (mm/sec)
W = seal contact width (mm)
Z_{cp} = fluid viscosity (N-sec/mm^2)

7

Measuring Seal Operating Characteristics

7.1 INTRODUCTION

Research and development of oil seals for end use applications can be a long process. It is difficult to determine if design or material changes actually result in performance increases in the field because field conditions are often unknown and they vary considerably. It is thus difficult to compare the performance of different designs and materials by testing in the field. Laboratory tests have been developed under carefully controlled conditions to evaluate new designs and materials and compare the results to determine if improvements have been made. These tests are described in the following sections. Results should be viewed with caution, as the final judgment is made on equipment operating in the field. Laboratory test results should always be substantiated with field test results.

7.2 MEASURING UNDERLIP TEMPERATURE

The oil seal is a friction device. Heat is generated by the contact of the shaft and the rubber lip and the shear of the fluid trapped in the contact zone. This heat causes the contact zone to be the hottest point in the sealing system. Knowledge of the underlip temperature and how it is affected by design and operating parameters will help prevent excessive heat from sludging oil and damaging the elastomer.

It is difficult to measure underlip temperature. Early researchers attempted to embed tiny thermocouples in the shaft, but this proved to be expensive because the delicate thermocouples and wires would break easily. The most common method of measuring underlip temperature is to focus an infrared camera with a very small focusing spot on the inside of a hollow steel shaft (Fig. 7.1). The camera scans the inside of the hollow shaft in the axial direction until the peak temperature is obtained. The peak temperature is directly opposite the sealing lip. The procedure will not give the actual underlip temperature, but a correction factor for the temperature drop across the hollow wall can be applied. The procedure is excellent when comparison of materials and designs is desired. If the wall is too thin, then the readings may not correlate with the heat conduction conditions that exist if the shaft is solid. Studies by Horve [69] indicate that a wall thickness of 1.500 mm (0.060 in.) will approach the heat transfer characteristics of a solid shaft. Sophisticated cameras and computerized systems (Fig. 7.2) can give a map of the entire seal region in real time. The underlip temperature at a local area can be observed to increase with each revolution as lobes on the shaft push the lip out, which increases the local load and thus the temperature. As the lobe passes, the load decreases and the local temperature also decreases.

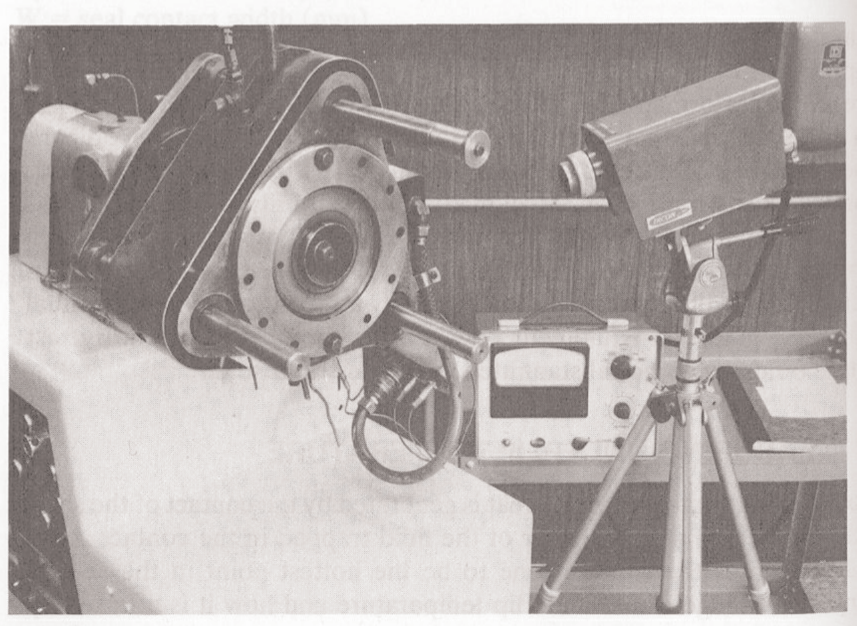


Fig. 7.1 Infrared camera can estimate values of underlip temperature. (Courtesy of Chicago Rawhide.)

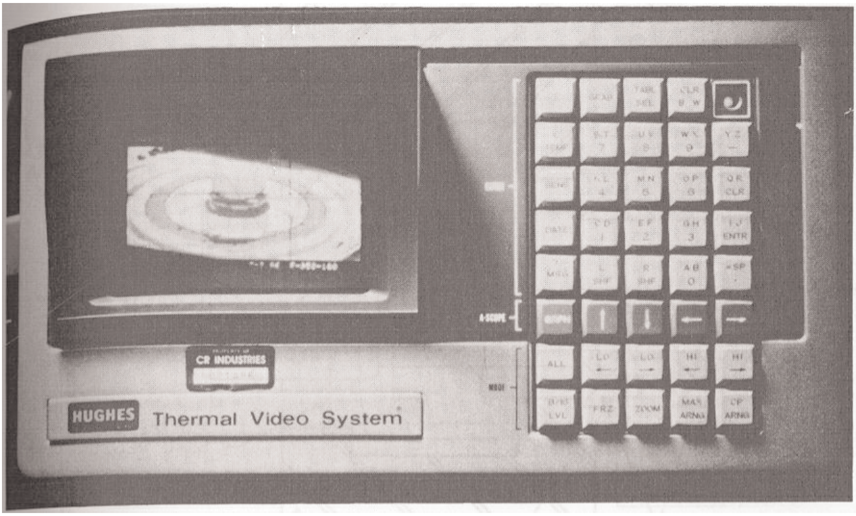


Fig. 7.2 Computerized infrared systems map the entire seal. (Courtesy of Chicago Rawhide.)

Attempts have been made to analytically estimate the underlip temperature. Early work by Upper [226] proved that the lip contact region was the hottest area. Many researchers now use finite-element analysis to map the seal region and predict lip temperature dependence on sump fill level and other heat conduction and convection characteristics [121, 224, 229].

7.3 MEASURING SEAL TORQUE AND POWER

The interference fit between the seal lip and the shaft generates a force that results in frictional torque and consumes energy as the shaft rotates. The frictional torque also generates heat, which can be detrimental to the seal element, the bearings, and the lubricant. Consistent methods of measuring seal torque and calculating power consumption are useful when comparing seal designs and materials in an attempt to reduce power loss without affecting seal performance [164, 202]. Seal function depends on maintaining a sealing force between the lip and the rotating shaft. If the load is reduced too much when the designer is attempting to reduce power consumption, then the seal may leak prematurely [44].

A crude measurement of seal torque can be made with a fixture to hold the seal and a torque wrench (Fig. 7.3). The fixture must minimize the shaft-to-bore misalignment (STBM), and the shaft dynamic runout (DRO) readings must be less than 0.08 mm (0.003 in.). The seal lip should be lightly

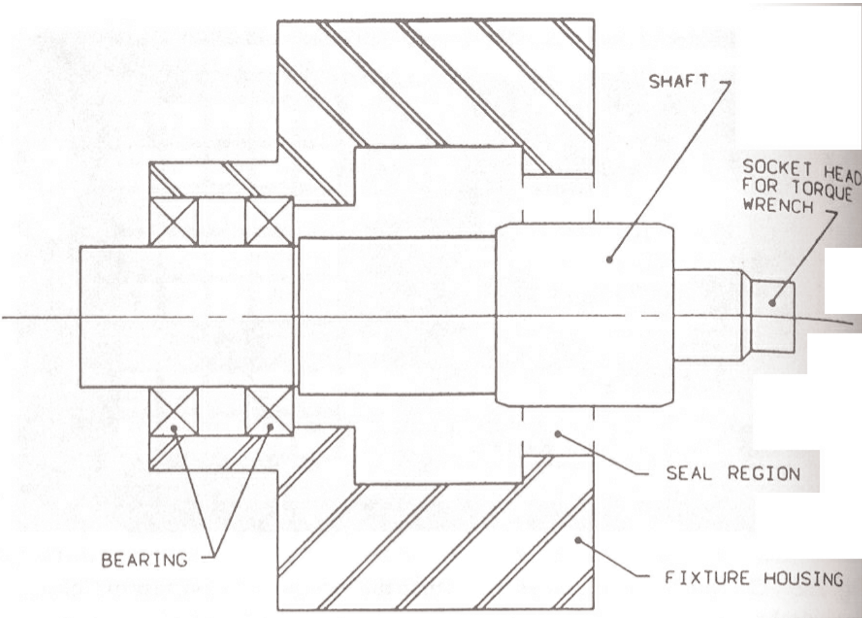


Fig. 7.3 Fixture to measure seal torque with torque wrench. (Courtesy of Rubber Manufacturers Association.)

lubricated. A torque wrench divided into 0.01 N-m (1.43 in.-oz) increments or smaller is placed on the end of the test shaft and slowly turned (60 rpm). The amount of torque can be read directly from the torque wrench as it is rotated. This method is only for guidance and will not predict the behavior of a seal under actual operating conditions. An electronic transducer placed between a drive motor and a typical oil seal test head will provide torque values at typical operating conditions (Fig. 7.4). Air bearings are often used to minimize system friction. System friction should be less than 1% of the measured seal torque. The depth to which the shaft is inserted through the seal into the fluid should be standardized. The volume of fluid contained within the test machine oil sump and the load on the shaft should also be standardized. Fluid drag on the shaft and housing can affect torque readings and lead to erroneous results if conditions vary from one test to another.

After the fluid has been heated to the desired temperature and the seal has been broken in for 20 hr at a speed of about 9 m/sec (about 1700 ft/min), then the torque can be recorded. The torque is typically measured at various shaft speeds and a chart of torque vs. speed is generated. The following conditions should be specified for each torque-speed curve: shaft speed, sump temperature, fill level, oil type, oil viscosity versus tempera-

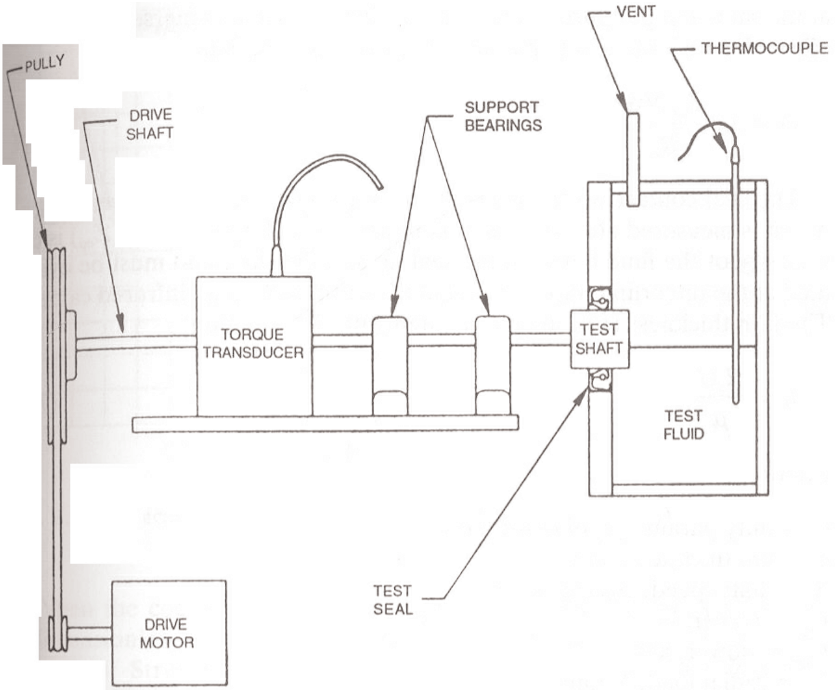

VENT
THERMOCOUPLE
PULLY
DRIVE SHAFT
SUPPORT BEARINGS
TORQUE TRANSDUCER
TEST SHAFT
TEST FLUID
TEST SEAL
DRIVE MOTOR

Fig. 7.4 Electronic torque-measuring machine. (Courtesy of Rubber Manufacturers Association.)

ture chart, shaft-to-bore misalignment, dynamic runout, angular misalignment (seal cock), the direction of shaft rotation, and whether any oil is present on the air side of the seal (i.e., when a pump test is under way). The power generated can be calculated from the frictional torque:

$$P_{\text{kW}} = 6.276 \times 10^{-3} \times \text{Tor} \times N \tag{7.1}$$

$$P_{\text{hp}} = 5.95 \times 10^{-5} \times \text{Tor}^{*} \times N \tag{7.2}$$

$$P_{\text{hp}} = 1.341\ P_{\text{kW}} \tag{7.3}$$

The coefficient of friction (μ) can be calculated if the frictional torque and the radial load are known.

$$\mu = \frac{\text{Tor}}{2\pi R_{\text{s}}^{2} R_{\text{L}}} \tag{7.4}$$

The radial load and the torque should be measured simultaneously. This is usually not practical, and radial load values are often measured after test at the same temperature as the sump during test while immersed in the test oil. A dimensionless duty parameter G can also be calculated.

The seal contact width (W) is the average value of the seal wear track, which is measured after the test is complete. The fluid viscosity (Zcp) is the viscosity of the fluid between the seal lip and the shaft and must be evaluated at the underlip temperature that is measured with an infrared camera. The film thickness (h) can be estimated with the equation.

$$G = 2\pi \frac{Z_{cp} NW}{R_L} \tag{7.5}$$

where

$$h = \frac{R_s G}{\mu} \tag{7.6}$$

G = duty parameter, dimensionless
h = film thickness, mm
N = shaft speed, rps (rev/sec)
P_{hp} = power, hp
P_{kW} = power, kW
R_L = radial load, N/mm
R_s = shaft radius, mm Tor = torque,
N-m Tor* = torque, in.-oz W =
seal contact width, mm Zcp = fluid
viscosity, N-sec/mm^2
μ = coefficient of friction, dimensionless

Fluid viscosity is often given in poise (P) or centipoise (cP). The following relationships are useful (dyn = dyne; lbf = pound force)

$$1 \text{ P} = 100 \text{ cP} = \frac{1 \text{ dyn-sec}}{\text{cm}^2} = 10^{-7} \frac{\text{N-sec}}{\text{mm}^2}$$

$$1 \text{ P} = 0.00209 \frac{\text{lb}_f\text{-sec}}{\text{ft}^2} = 1.45 \times 10^{-5} \frac{\text{lb}_f\text{-sec}}{\text{in.}^2}$$

$$1 \text{ cP} = 0.01 \text{ P} = \frac{0.01 \text{ dyn-sec}}{\text{cm}^2} = 10^{-9} \frac{\text{N-sec}}{\text{mm}^2}$$

$$1 \text{ cP} = 2.09 \times 10^{-5} \frac{\text{lb}_f\text{-sec}}{\text{ft}^2} = 1.45 \times 10^{-7} \frac{\text{lb}_f\text{-sec}}{\text{in.}^2}$$

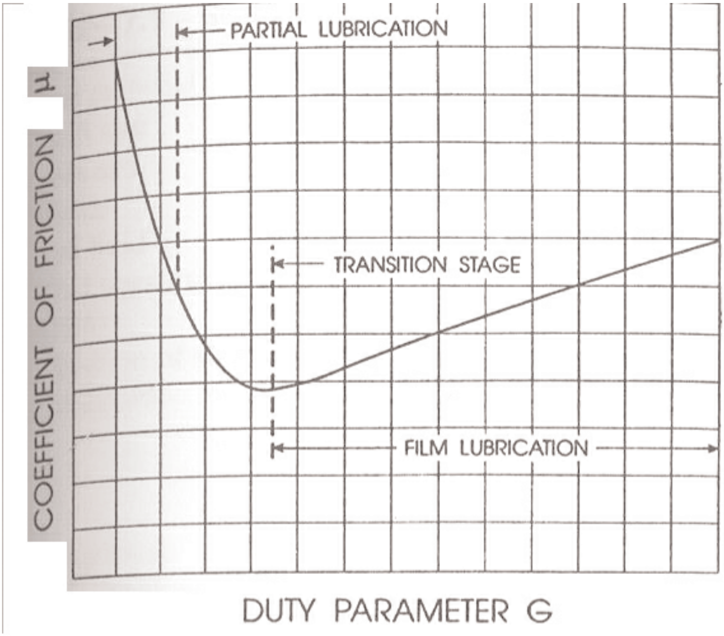


Fig. 7.5 Topical Stribeck curve. (Courtesy of Chicago Rawhide.)

When the coefficient of friction is plotted against the duty parameter, a dimensionless curve is generated that is known as a Stribeck curve (Fig. 7.5). Stribeck curves are often used to investigate the frictional behavior of bearings, machinery, seals, and other components. At low values of G, the coefficient of friction is large and decreases as G increases. This region corresponds to a poorly lubricated or boundary condition with some physical contact between the surfaces of the components. At large values of G, the coefficient of friction may increase with G. This represents the region of full lubrication with no contact between components. The friction generated is due to shear in the fluid film. The region between the two extremes is where the coefficient of friction reaches a minimum and is referred to as the mixed film lubrication regime (Fig. 7.6).

Example. A seal for a 76.2 mm (3.000 in.) shaft made with a fluoroelastomer lip is being analyzed. The seal was broken in at 2000 rpm and 150°C (302°F) for 20 hr, and a torque versus speed curve was generated for sump temperatures of 149°C (300°F), 121°C (250°F), and 93°C (200°F). The underlip temperature was measured continuously as the shaft speed increased. After test, the hot radial load was measured at the three temperatures, and the wear track width was measured at room temperature.

The seal data are given in Table 7.1. The measured data and calculations appear in Tables 7.2-7.4. At low speeds, the measured underlip

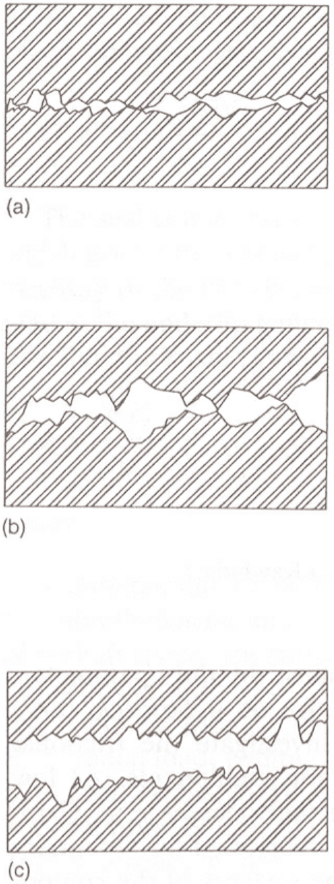


Fig. 7.6 Lubrication regimes. (a) Boundary lubrication with surfaces in contact.(b) Mixed film lubrication has some contact between surfaces that are partially lubricated with a film of oil. (c) Fluid film lubrication has separated the surfaces with a bulk lubricant film. (Courtesy of Chicago Rawhide.)

temperature is less than the sump temperature because of the temperature drop across the thin wall of the hollow shaft. The measured underlip temperature was corrected to account for this temperature drop. The oil viscosity shown in the tables was determined using the viscosity temperature data in Fig. 7.7 for Mobil 1 oil. The corrected underlip temperature was used to determine the viscosity at each speed. Examination of the data in Tables 7.2-7.4 shows that the underlip temperature, torque, and power consumption all increase as the shaft speed increases. The underlip fluid viscosity decreases with increasing shaft speed. The Stribeck curve (Fig. 7.8) demonstrates that radial lip seals operate in various lubrication regimes. The curve reaches a minimum that indicates that the seal moves

Table 7.1 Seal Data

Seal material	Fluoroelastomer
Shaft size	76.2 mm (3.000 in.)
Bore size	101.6 mm (4.000 in.)
Width	12.7 mm (0.500 in.)
Test fluid	Mobil 1
Fluid pressure	0.0
STBM[a]	
Direction of shaftrotation	Clockwise
Procedure:	Run 20 hourbreak-in at 2000 rpm and 149° C (300° F). Measure torque and sump temperature at lowest possible speed. Increase in 100 rpm increments to 500 rpm. Increase in 500 rpm increments after that to highest speed.
Sump level	Full
Wear track width	0.36 mm (0.014 in.)
Radial load in hot oil;	at 93° C (200° F) 0.102 N/mm (9.4 oz/in.)
	at 121° C (250° F) 0.0985 N/mm (9.0 oz/in.)
	at 149° C (300° F) 0.0854 N/mm (7.8 oz/in.)

[a] STBM = Shaft-to-bore misalignment.

Source: Chicago Rawhide.

from an unlubricated state to a lubricated state as the shaft speed increases.

The coefficient of friction is quite high for lubricated components because the film is very thin. The film thickness increases with shaft speed and reaches a maximum of about 0.38-0.40 μm (about 15 μin.) (Fig. 7.9). This maximum is the same magnitude as the original surface finish of the shaft. Fig. 7.9 Film thickness (μm) versus duty parameter G. (Courtesy of Chicago Rawhide.)

Ishiwata [94] used the Stribeck curve to analyze the performance of seals over a range of test conditions. The torque was measured at the operating conditions, the radial load was measured at room temperature, the fluid viscosity was calculated at sump temperature, and the contact width was measured on a glass shaft. Ishiwata was unable to measure underlip temperature during the tests. The data were used to calculate the coefficient of friction and the duty parameter G. Some of the seals leaked during measurement, and their values of torque and friction were lower than those of seals that did not leak (Fig. 7.10). Ishiwata speculated that a sealing limit exists and that any value of μ falling below the curve given by Equation (7.7) would be the result of a leaky seal. Other researchers dispute the existence of a sealing limit and suggest that the results may change dramatically if the radial load measurements are made at sump temperature and the fluid viscosity is determined at the actual underlip temperature.

$$\mu = 30.42G^{0.341} \qquad (7.7)$$

Table 7.2 Seal Characteristics at 93° C (200° F)

Shaft speed (rpm)	Measured torque (in.-oz)	Adjusted underlip temperature (° F)	Underlip fluid viscosity (cP)	Power consumption (Ws)	Coefficient of friction μ	Duty parameter G	Film thickness h (μm)
50	39.5	202	9.46	1.46	0.297	0.171×10^{-6}	0.0219
100	37.5	204	9.25	2.77	0.282	0.335	0.0453
200	38.0	210	8.65	5.62	0.286	0.626	0.0834
300	40.0	220	7.78	8.87	0.301	0.845	0.107
400	41.0	226	7.31	12.11	0.309	1.058	0.130
500	43.0	228	7.17	15.9	0.324	1.298	0.153
1000	52.0	237	6.56	38.5	0.391	2.37	0.231
1500	59.0	247	5.97	65.4	0.444	3.24	0.278
2000	64.5	258	5.40	95.4	0.485	3.91	0.307
2500	70.0	270	4.87	129.4	0.527	4.41	0.319
3000	76.0	275	4.67	168.6	0.572	5.07	0.367
3500	81.5	280	4.48	210.9	0.613	5.68	0.378
4000	86.5	285	4.30	255.9	0.651	6.23	0.359
4500	92.5	289	4.17	307.8	0.696	6.79	0.372
5000	96.5	294	4.00	356.8	0.726	7.24	0.380
5500	103.5	297	3.91	421.0	0.779	7.78	0.380

Source: Chicago Rawhide.

Table 7.3 Seal Characteristics at 121° C

Shaft speed (rpm)	Measured torque (in.-oz)	Adjusted underlip temperature (° F)	Underlip fluid viscosity (cP)	Power consumption (Ws)	Coefficient of friction μ	Duty parameter G	Film thickness h (μm)
50	44.5	250	5.80	1.65	0.350	0.109 x10^{-6}	0.0119
100	41.0	253	5.65	3.03	0.322	0.214	0.0253
200	38.5	258	5.40	5.69	0.303	0.430	0.0541
300	37.5	268	4.95	8.32	0.295	0.561	0.0725
400	38.0	272	4.79	11.2	0.299	0.724	0.0923
500	38.5	274	4.71	14.2	0.303	0.890	0.112
1000	45.0	284	4.34	33.3	0.354	1.640	0.177
1500	51.5	290	4.13	57.1	0.405	2.34	0.220
2000	56.0	296	3.94	82.8	0.440	2.98	0.258
2500	61.0	303	3.74	112.8	0.479	3.53	0.281
3000	65.5	308	3.60	145.3	0.515	4.08	0.302
3500	70.0	314	3.45	169.5	0.550	4.56	0.316
4000	75.5	320	3.30	223.0	0.593	4.99	0.321
4500	80.0	323	3.23	266.2	0.629	5.49	0.333
5000	86.0	326	3.16	318.0	0.676	5.97	0.336
5500	91.0	329	3.10	370.1	0.715	6.44	0.343

Source: Chicago Rawhide.

Table 7.4 Seal Characteristics at 149° C (300° F)

Shaft speed (rpm)	Measured torque (in.-oz)	Adjusted underlip temperature (° F)	Underlip fluid viscosity (cP)	Power consumption (Ws)	Coefficient of friction μ	Duty parameter G	Film thickness h (μm)
50	55.0	296	3.94	2.03	0.499	0.0858 x 10^{-6}	0.00655
100	51.0	301	3.80	3.77	0.462	0.166	0.0137
200	47.0	308	3.60	6.95	0.426	0.314	0.0281
300	45.0	318	3.35	9.98	0.408	0.438	0.0409
400	44.0	327	3.14	13.01	0.399	0.548	0.0523
500	44.0	330	3.08	16.3	0.399	0.671	0.0641
1000	46.0	336	2.95	34.0	0.417	1.28	0.117
1500	50.0	342	2.83	55.5	0.453	1.85	0.156
2000	53.0	350	2.69	78.4	0.481	2.35	0.186
2500	57.0	356	2.58	105.4	0.517	2.81	0.207
3000	61.5	363	2.47	136.2	0.558	3.23	0.221
3500	65.0	369	2.38	168.2	0.589	3.63	0.235
4000	70.0	374	2.31	207.1	0.635	4.03	0.242
4500	74.5	378	2.25	247.9	0.676	4.41	0.249
5000	79.0	382	2.20	292.1	0.716	4.80	0.255
5500	83.0	384	2.17	337.6	0.753	5.20	0.263

Source: Chicago Rawhide.

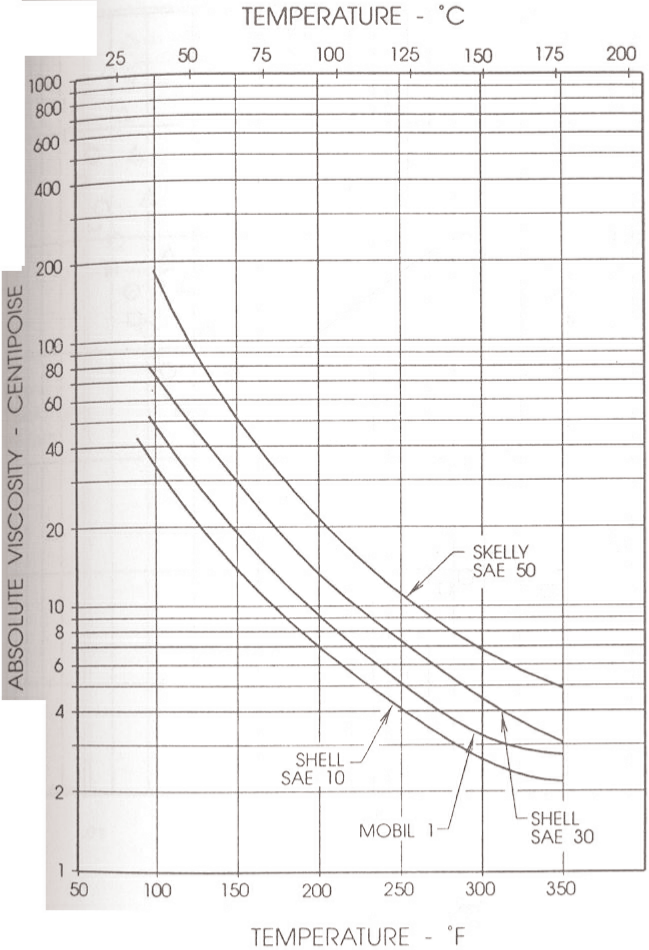


Fig. 7.7 Absolute viscosity (Zcp) versus temperature. (Courtesy of Chicago Rawhide.)

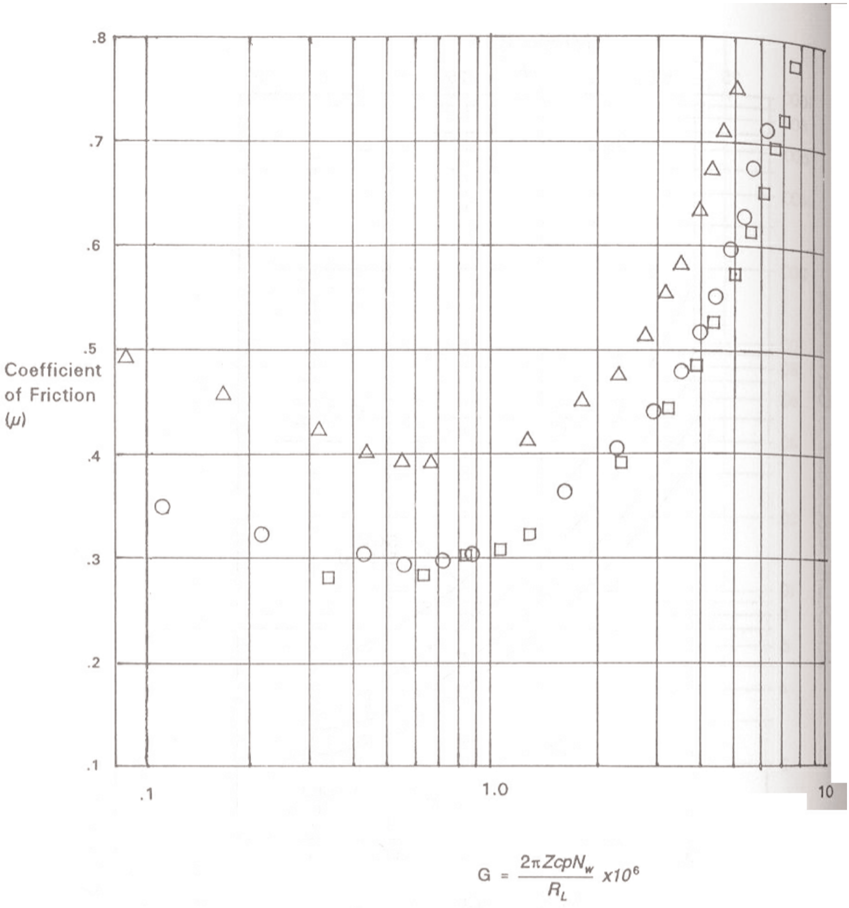


Fig. 7.8 Stribeck curve. (Δ) 149° C (300° F); (○) 121° C (250° F); (□) 93° C (200° F). (Courtesy of Chicago Rawhide.)

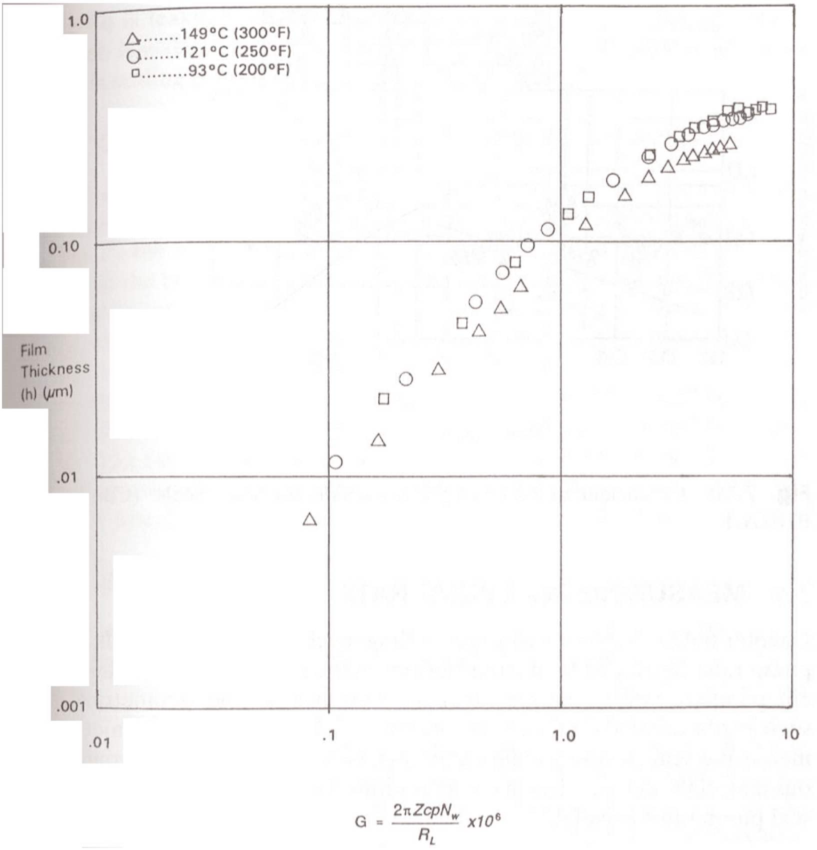


Fig. 7.9 Film thickness (μm) versus duty parameter G. (Courtesy of Chicago Rawhide.)

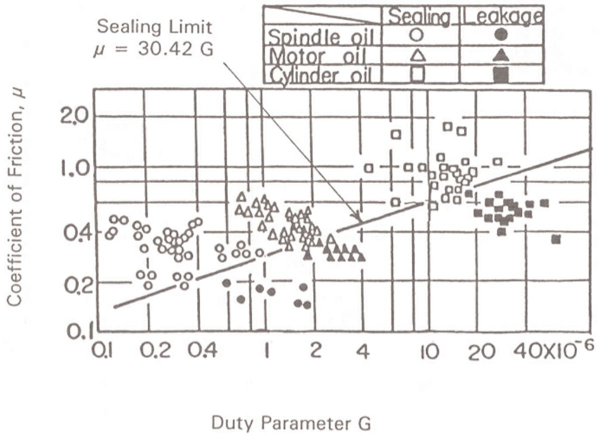


Fig. 7.10 Coefficient of friction of leaky versus nonleaky seals. (Courtesy of BHRA.)

7.4 MEASURING SEAL PUMP RATE

Chapter 6 describes a microscopic sealing mechanism that depends upon a pump rate developed by microasperities in the seal wear track. This inherent pumping ability can be augmented by macroscopic geometrical aids such as ribs molded on the air side of the seal. It is obvious that a method of measuring seal pumping ability is necessary to evaluate and compare various materials and seal design configurations. Several methods of measuring seal pump rates exist [196].

7.4.1 Spiral Groove Shaft

A shaft is prepared with a spiral groove cut or etched into the surface with a helix angle of 45° . The groove is 0.02-0.08 mm (0.0008-0.0031 in.) deep with a width of about 0.8 mm (0.031 in.). The shaft is rotated in a direction that allows the spiral groove to auger or pump oil from the oil side to the air side, which causes a leak. The leakage rates of various seal configurations can be measured and compared. The seal with the lowest leak rate will have the highest pump rate that retards the flow. A sealing capacity (Sc) can be defined as the difference between the leakage rates of a seal design with and without hydrodynamic aids (Equation 7.8).

$$\mathrm{Sc} = 100\left(1 - \frac{\mathrm{Leh}}{\mathrm{Le}}\right) \tag{7.8}$$

where Le is leakage rate (cm^3/min) of seal design without hydrodynamic aids, Leh is leakage rate (cm^3/min) of seal design with hydrodynamic aids, and Sc is sealing capacity.

7.4.2 Other Methods of Estimating Sealing Capacity

Several other methods of estimating the effectiveness of hydrodynamic seals have been reported. A section of the seal lip is carefully removed with a jeweler's file or a razor blade, and the seal is installed with the damaged portion at the bottom of the shaft. The leakage rates are measured for seals with and without hydrodynamic aids, and the sealing capacity is determined with Equation (7.8). Test results can be erratic because it is very difficult to reproduce nicks in seal lips.

Another method involves grinding a series of evenly spaced flat spots about 0.04x0.08 mm deep on the circumference of the shaft and determining the leak rates of seals with and without hydrodynamic aids. This test is limited because the flats are more severe than conditions found in the field and may interfere with the proper functioning of the hydrodynamic ribs.

7.4.3 Backward Test

When a seal is installed backward (i.e., the air side is facing the oil sump), instantaneous leakage will occur (Chapter 6). This phenomenon can be used to determine the sealing capacity of a hydrodynamic design. The leakage rate of a seal without hydrodynamic aids is determined and compared to the leakage rate of a seal with the hydrodynamic aids and the same cross section and material. It is assumed that the hydrodynamic aids will provide an additional pumping effect to increase the leakage rate of the reversed seal geometry. The sealing capacity for the backward test is calculated using the equation

$$\mathrm{Sc_B} = 100\left(1 - \frac{\mathrm{Le_B}}{\mathrm{Leh_B}}\right) \tag{7.9}$$

where $\mathrm{Sc_B}$ is the sealing capacity using backward installation, LeB is the leakage rate of the seal installed backward without hydrodynamic aids (cm^3/min), and LehB is the leakage rate of the seal installed backward with hydrodynamic aids (cm^3/min). Seals with damaged lips or any other shaft or seal problems may increase leakage and give the impression that the pumping aids are more effective than they actually are. This method does not simulate conditions that occur in the actual applications because users do not normally install seals backward. The method does not accurately simulate the oil transfer that occurs in the actual application because the seal hydrodynamic aids are not pumping against a head of oil on the normal oil side of the lip. These limitation can lead to unreliable results and are not recommended for determining seal pumping ability even though some seal suppliers and users still use this method. The oil drop test is preferred.

7.4.4 Oil Drop Test

The oil drop test is performed by mounting the seal in the normal position on either a horizontal or vertical shaft and injecting a small quantity (0.02 mL) of oil treated with a fluorescent dye into the space between the rotating shaft and the seal barrel angle at a point near the contact area. If the shaft is horizontal, the oil droplet should be injected near the top of the seal. It is often difficult to ensure that the oil droplet is uniformly distributed around the shaft when the shaft is horizontal. The recommended method uses a horizontal seal with a vertical shaft (Figs. 7.11 and 7.12). If the seal has a secondary lip, it is trimmed away. The sump is completely filled with oil to a consistent level above the seal location to eliminate the effects of gravity and ensure the presence of a constant head of oil. Seals can pump air into the

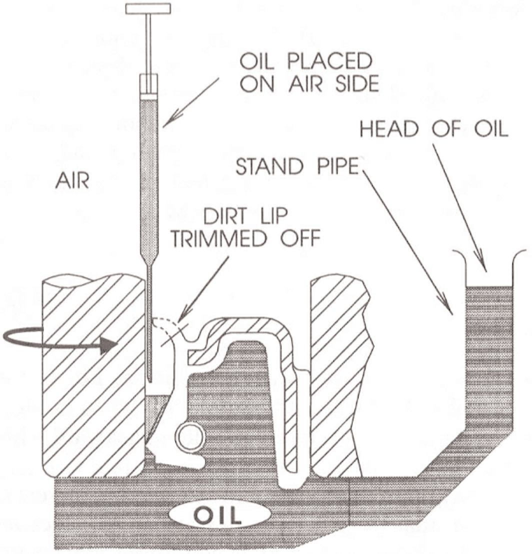


Fig. 7.11 Measurement of seal pump rate. (Courtesy of Chicago Rawhide.)

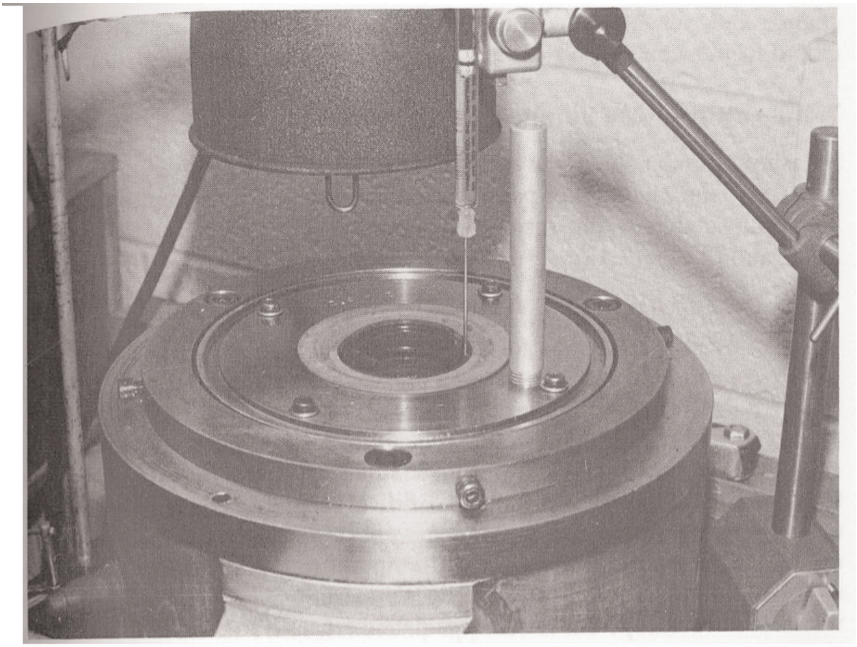


Fig. 7.12 Equipment to measure seal pump rate.

sump, and an air bubble may form under the seal [38]. After the shaft has stopped rotating, the seal lip should be carefully lifted periodically to allow the air to escape. The seal is normally run 20 hr while sealing a head of oil to allow the seal lip to establish a wear track on the rotating shaft. After the break-in period, the small drop of fluorescent-dyed oil is injected at the air side of the seal. The pump time is obtained with a stop watch by observing (under ultraviolet light) when the fluorescent oil meniscus on the air side of the seal disappears. The pump rate Q is calculated as

$$Q = \frac{V_{\mathrm{oil}}}{\mathrm{PT}} \qquad \mathrm{cm}^3/\mathrm{sec} \tag{7.10}$$

where Voil is the oil droplet volume (cm^3) and PT is the pumping time (sec).

If seal torque is measured simultaneously with pump rate and underlip temperature, then analysis of the torque versus time data can provide an accurate pumping time. The seal torque changes instantaneously when the oil is dropped on the air side and returns to the original value after the meniscus disappears. The electronics can be programmed to detect this change and calculate a pumping time, or the pumping time can be estimated from a chart recorder.

7.4.5 Pump Rate versus Duty Parameter

Measured pump rates (Q) vary with shaft speed, sump temperature, seal geometry, and many other parameters (Chapter 8). These pump rates can be normalized to a dimensionless number (Q*) using Equations (7.5), (7.6), and (7.11)-(7.13).

$$Q_{\text{norm}} = 2\pi R_s h V_s \mu \quad \text{mm}^3/\text{sec} \tag{7.11}$$

$$V_s = 2\pi R_s N \quad \text{mm/sec} \tag{7.12}$$

$$Q^* = \frac{Q}{Q_{\text{norm}}} = \frac{Q}{4\pi^2 R_s^3 G N} \tag{7.13}$$

A series of measurements was made at different speeds, sump temperatures, and shaft sizes and with two different seal designs. The torque, underlip temperature, and pump rate were measured simultaneously. The radial load was measured at sump temperature, the fluid viscosity was evaluated at the measured underlip temperature, and the seal contact width was measured after test. These data were used to calculate the coefficient of friction μ (Equation 7.4), the duty parameter G (Equation 7.5), and Q* (Equation 7.13). When the coefficient of friction is plotted against

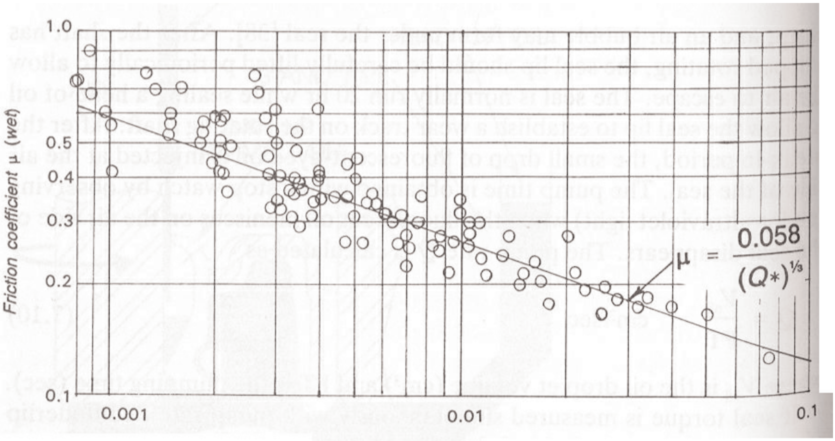


Fig. 7.13 Friction coefficient versus Q* . (Courtesy of Chicago Rawhide.)

Q* (Fig. 7.13), many of the data collapse into a band that can be represented by Equation (7.14). Equation (7.15) results when (7.13) is substituted into (7.14).

$$\mu = \frac{0.058}{(Q^*)^{1/3}} \tag{7.14}$$

$$\mu = 0.197R_s\left[\frac{GN}{Q}\right]^{1/3} \tag{7.15}$$

The ratio of shaft speed to seal pump rate (N/Q) is approximately constant with speed; therefore, the coefficient of friction is directly proportional to shaft size and to $G^{1/3}$. It is interesting to note that the exponent (0.3333) of G in Equation (7.15) is similar to the exponent 0.341 reported by Ishiwata in Equation (7.7). The relationship between coefficient of friction and pump rate suggests that the fluid film between the seal lip and the shaft changes when seal pump rate changes. Pump rate also increases as shaft speed increases. These phenomena may all be interrelated. More studies are required. A plot of normalized pump rate versus duty parameter appears in Fig. 7.14. The normalized pump rate decreases as the duty parameter increases. This relationship appears in the equation

$$Q^* = \frac{2.632 \times 10^{-10}}{G^{1.33}} \tag{7.16}$$

7.5 MEASURING SEAL LIFE

7.5.1 Laboratory Machines for Endurance Testing

Seal life is very difficult to predict or measure on equipment operating in the field. There are many variables, and these variables are constantly changing. Application variables that affect performance are listed in Table 7.5. These field variables interact with seal variables (Table 7.6), and leakage can occur. Identifying the cause of leakage is a difficult task because the number of variables and their unpredictable changes can mask the root problems. It is also difficult to evaluate new materials and designs and determine through field tests alone whether improvements over existing materials and designs have been accomplished.

Seals are often tested on laboratory machines with which application variables can be stimulated, closely controlled, and monitored [21, 26, 31, 53, 158, 161, 167, 188, 196]. Specifications for test machines are given in Table 7.7. A schematic of a typical test head is shown in Fig. 7.15, and a typical seal functional test machine appears in Fig. 7.16.

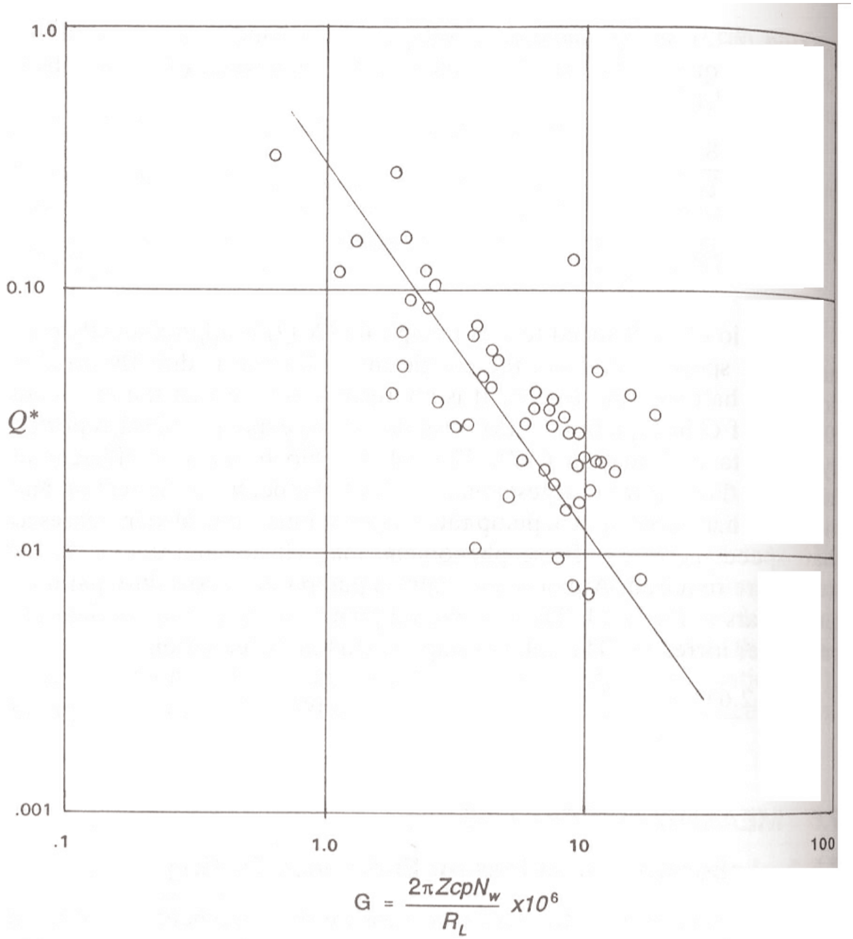


Fig. 7.14 Normalized pump rate Q* versus duty parameter G. (Courtesy of Chicago Rawhide.)

The test head temperature is controlled by circulating a heat transfer fluid through a chamber in the test head that is isolated from the test fluid or by using electric heaters. In some cases, a combination of the two is employed. The shaft speed is carefully controlled, and the spindle bearings are chosen to minimize shaft whip and vibrations. The plates that the seals are installed in can be adjusted to provide a known and controllable shaft-to-bore misalignment. The test shaft is ground to specifications and can be adjusted to provide the desired dynamic runout. It is recommended that

Table 7.5 Application Variables that Affect Seal Performance

Ambient

1. Temperature and range
2. Ozone
    a. Ultraviolet
    b. Infrared
    c. Other chemical activity
3. Dust

Medium

1. Viscosity
2. Viscosity index
3. Temperature
4. Pressure load
5. Chemical activity
6. Breakdown products
7. Contaminants

Shaft

1. Speed
2. Eccentricity
3. Diameter
4. Finish
5. Axial movement
6. Material

Housing

1. Finish
2. Eccentricity
3. Interference tolerance

Operation

1. Cycles per day
2. Run hours per cycle
3. Down time

Source: Chicago Rawhide.

seals be run until failure occurs and that Weibull statistics be used to analyze the results and compare the performance of various seal types and materials. The definition of failure must be established and used consistently throughout the series of tests.

The seal supplier and user must agree on the definition of failure. Possible definitions include the first drop of oil that passes the lip, oil flowing down the face plate, or a prescribed quantity of oil that accumulates during a specific time period. Many different failure mechanisms exist. Seal and shaft damage, poor seal design, poor material composition, manufacturing defects, and test machine failures all contribute to early leakage. Bench tests may be run using accelerated application conditions. These conditions typically include high temperatures, excessive shaft speed, and exaggerated shaft dynamic runouts that exceed application conditions. Care must be taken to choose test conditions that do not exceed seal material, seal de-

Table 7.6 Seal Variables that Affect Seal Performance

Material	Design
1. Oil swell	1.Radial load
2. Stress relaxation	a. Spring tension
3. Creep	b. Element hoop load
4. Thermal effects	c. Element beam load
a. Expansion	d. Contact area
b. Inversion	
5. Hysteresis and resilience	2. Overall configuration
6. Strain rate sensitivity	a. R value
7. Viscoelastic vibration and damping	b. Number of lips
8. Crystallization	c. Air-side angle
9. Cross-linking	d. Oil-side angle
10. Frictional properties	e. Volume of rubber
11. Electrical effects	f. Element length
12. Hardness	g. Stamping size and material
13. Abrasion resistance	
14. Corrosion rating	3. Manufacturing tolerance and practices
15. Chemical stability	4. Cost

Source: Chicago Rawhide.

Table 7.7 Endurance Test Machine Specifications

Test head	Level, rigid, vented to atmosphere
Spindle speed control	±3%
Spindle runout	±0.03 mm (±0.001 in.)
Up to 6000 rpm	
Test shaft-to-bore alignment	±0.03 mm (±0.001 in.)
Test head temperature control	±3° C (±5° F)[a]

[a] Through the operating temperature range.

Source: RMA(OS-12).

sign, or test fluid capabilities. The list of failure modes is almost endless, and the proper statistical method should be employed to analyze the results. The statistical analysis should be coupled with a careful review of the seal, test fluid, test shaft, and housing to ensure that the cause of failure for each seal is known. The application variables should be monitored during the test and checked for accuracy before and after test. Seal variables should also be measured before and after test (Table 7.8).

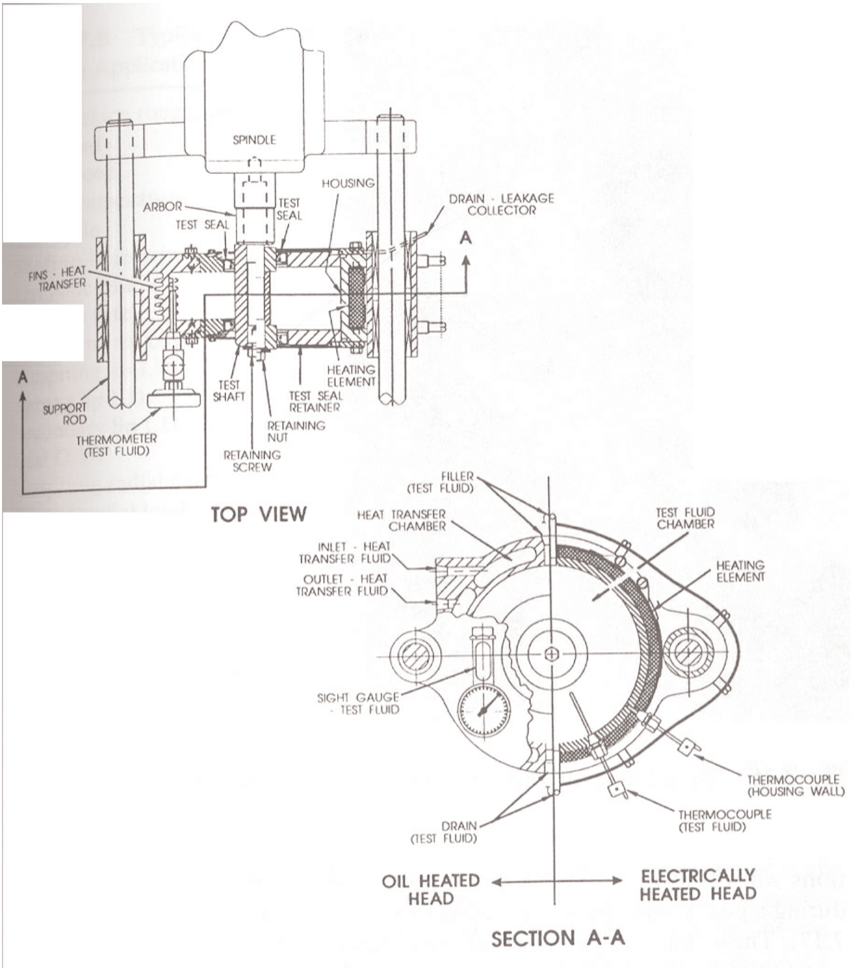

SPINDLE
HOUSING
ARBOR
TEST SEAL
TEST SEAL
DRAIN - LEAKAGE COLLECTOR
A
FINS - HEAT TRANSFER
A
HEATING ELEMENT
TEST SHAFT
TEST SEAL RETAINER
SUPPORT ROD
RETAINING NUT
THERMOMETER (TEST FLUID)
RETAINING SCREW
TOP VIEW
FILLER (TEST FLUID)
HEAT TRANSFER CHAMBER
TEST FLUID CHAMBER
INLET - HEAT TRANSFER FLUID
HEATING ELEMENT
OUTLET - HEAT TRANSFER FLUID
SIGHT GAUGE - TEST FLUID
THERMOCOUPLE (HOUSING WALL)
THERMOCOUPLE (TEST FLUID)
DRAIN (TEST FLUID)
OIL HEATED HEAD
ELECTRICALLY HEATED HEAD
SECTION A-A

Fig. 7.15 Schematic of oil seal test machine. (Courtesy of SAE.)

7.5.2 Histograms, Success Testing, and Reliability

Seals are often tested for a given period of time (100x1000 hr). If they do not leak during this time frame and meet predetermined wear criteria, then they are considered approved. This technique is called success testing and provides limited performance information at low confidence levels about the general seal population. As an example, consider 200 seals from a large population that have been tested. Material, process design, and test varia-

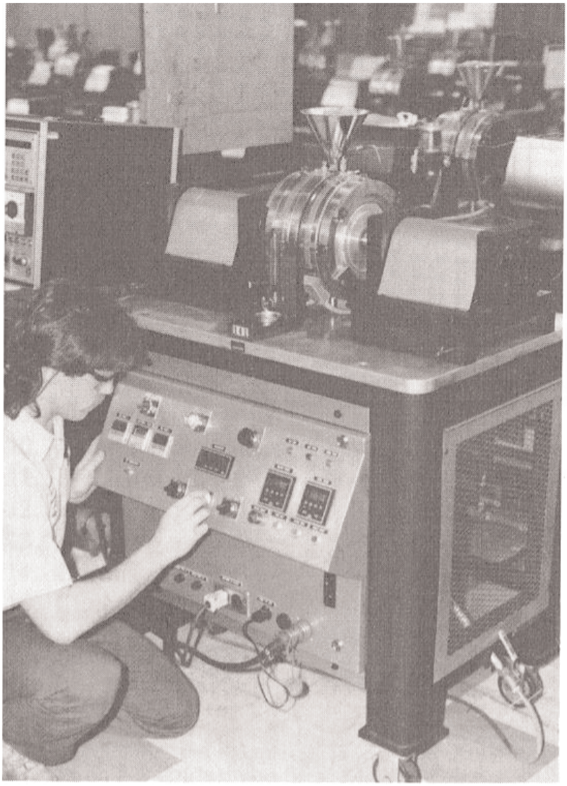


Fig. 7.16 Typical oil seal functional test machine. (Courtesy of Chicago Rawhide.)

tions will result in variations in the time to leak. The number of failures during a given time can be plotted versus time to provide a histogram (Fig.7.17). These data can be converted to a percent failure per hour or a failure rate (F) with the equation

$$F = \frac{100\ N_{\mathrm{F}}}{\Delta T\ N_{\mathrm{T}}} \tag{7.17}$$

A cumulative percent failure rate (FT) is calculated using the equation

$$F_{\mathrm{T}} = \frac{100 N_{\mathrm{FC}}}{N_{\mathrm{T}}} \tag{7.18}$$

In these equations, N_F is the number of failures during time frame ΔT, ΔT is the time increment (hr), N_T is the total number of seals tested, and N_{FC} is the cumulative number of failures.

Table 7.8 Typical Check List for Seal and Application Variables

Shaft surface roughness	MRB
Shaft lead	MRB
Shaft speed	M
Sump temperature	M
Test cycle	M
Shaft-to-bore misalignment	MRBA
Dynamic runout	MRBA
Leakage rate	M
Garter spring load	MRB
Unsprung lip I.D.	MRBA
Sprung lip I.D.	MRBA
Secondary lip I.D.	MRBA
Seal O.D.	MRB
Unsprung radial load	MRBA
Sprung radial load	MRBA
Unsprung LOP	MRBA
Sprung LOP	MRBA
Seal lip wear track width	MRA
Shaft groove width and depth	MRA

M = Monitored during test; MRBA = measured and recorded before and after test; MRB = measured and recorded before test; MRA = measured and recorded after test.

Source: Chicago Rawhide.

When the time frame (ΔT) becomes very small and the number of seals (N_T) tested becomes very large, then the percent failure rate and the cumulative percent failure rate will become continuous curves as shown by the dashed lines in Fig. 7.17 and 7.18. The failure rate distribution curve reflects the material, process, and design variations present in a given population of seals.

When a few seals are used with success testing to qualify seals, then it is difficult to determine if the general population meets the required reliability (Equation 7.19).Increasing the number of tests increases the probability that the desired reliability is achieved. The number of tests required without leakage to provide a given confidence level that a given reliability will be achieved is charted in Fig. 7.19. Analysis of this figure reveals that 165 tests must be run without leakage to give a confidence level of only 80% that 99% of the seal population will pass the approval test. If leakage occurs during the test program, then the confidence level drops dramati-

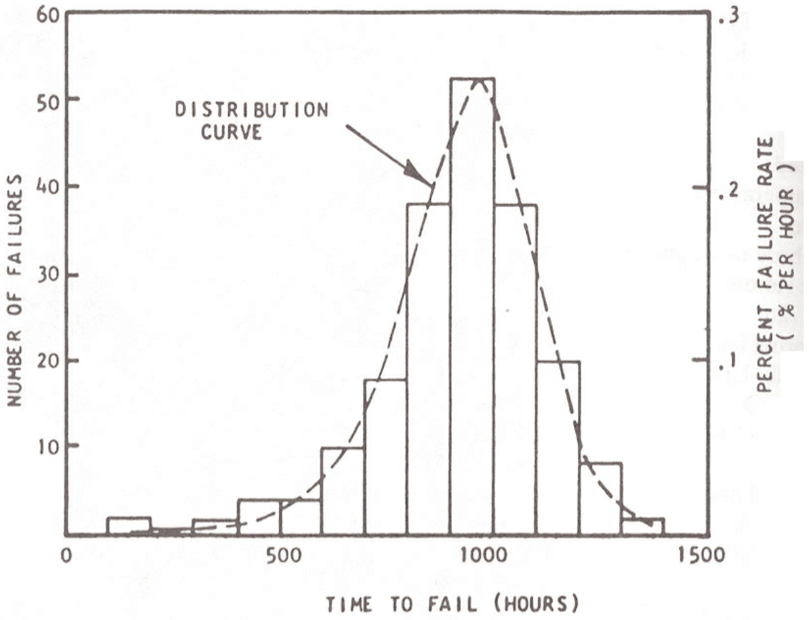


Fig. 7.17 Seal failure histogram.

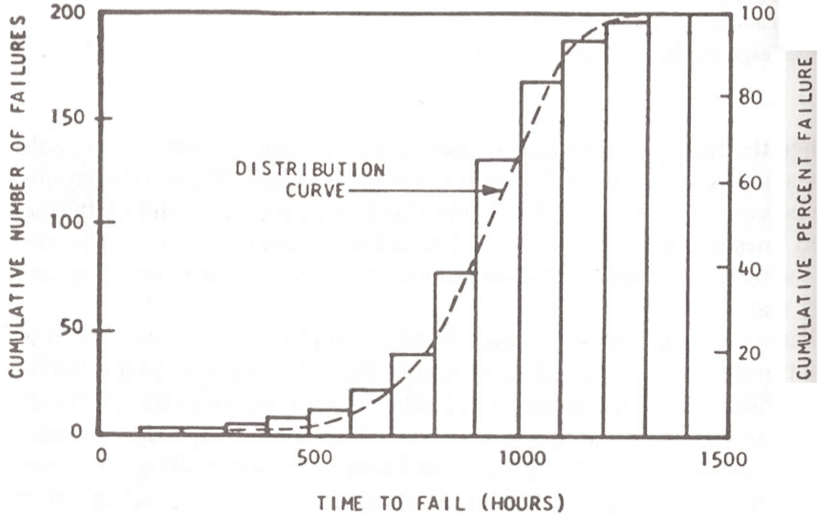


Fig.7.18 Cumulative number of failures and cumulative percent failure versus time to fail.

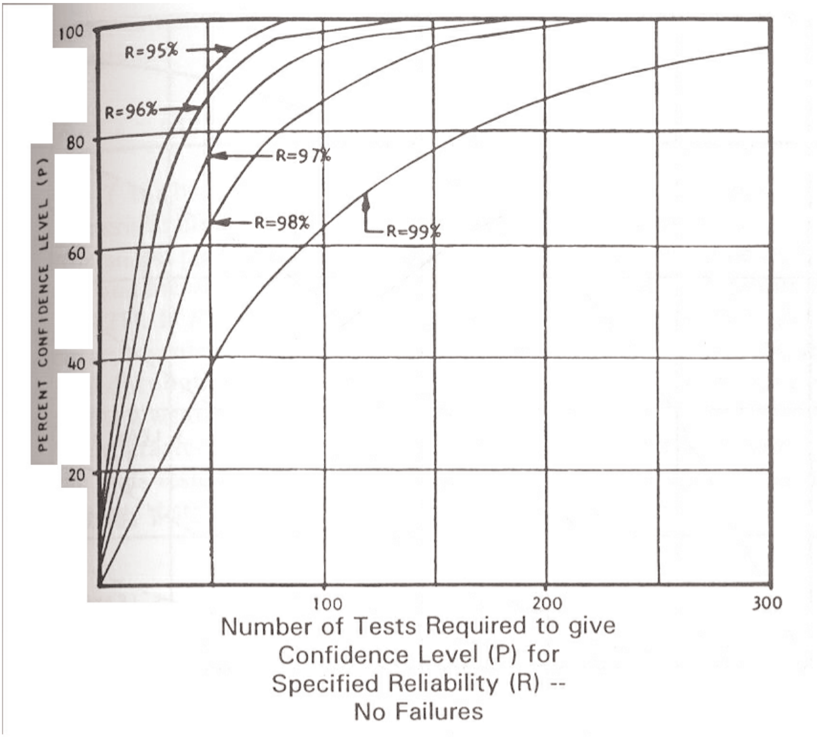


Fig. 7.19 Confidence chart for seal tests without leakage.

cally (Fig. 7.20). If one seal out of 165 leaks, then the confidence level that the population will have 99% reliability will drop to 45%.

One hundred thirty-five additional seals must be tested to move the confidence level back to 80%. Testing this quantity of seals to gain reasonable confidence levels is prohibitively expensive. If a reasonable number of seals (eight) are tested without failure, the confidence that 99% of the population will pass the qualification test is only 7%. Seal users and supplies must understand the limitations of success testing when planning test programs.

$$Re = 100 - L_{\mathrm{F}} \tag{7.19}$$

where Re is percent reliability and LF is the allowable percent failure of the population.

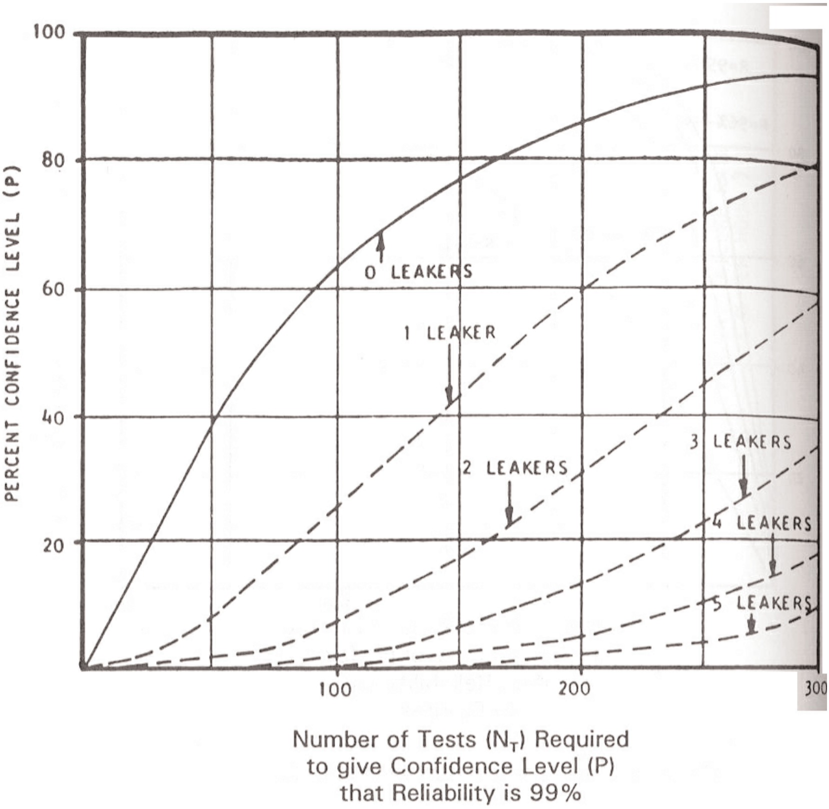


Fig. 7.20 Seal leakage during success test dramatically reduces confidence level.

7.5.3 Weibull Statistics for Seal Tests

The data appearing in Figs. 7.19 and 7.20 are based on a single failure mode with a normal distribution of leakers. This ideal condition is not found in actual field performance or in laboratory bench tests. Early leakers will have a high initial failure rate that declines rapidly to zero. A random failure rate declines slowly with time, and catastrophic failure distributions have all failures occurring within a short time frame. Seal test data contain combinations of these failure modes. The Weibull function is a single mathematical function that has the flexibility to adapt to this range of failure modes. The technique has been widely used in the electronics industry, the bearing industry, the seal industry [49, 73], and elsewhere. The function F_t, has three parameters that define the shape, location, and spread of the failure rate distribution.

$$F_T = 1 - \exp\left[-\left(\frac{T - T_0}{\theta}\right)^B\right] \qquad (7.20)$$

where T is observed life (hr), T_0 is a location parameter (hr), θ is the characteristic life (hr), and B is a dimensionless shape parameter.

The shape parameter or Weibull slope (B) defines the type of failure mode, and the value of B will determine the shape of the failure rate curve (Fig. 7.21). If B = 1, the failure rate curve is random. If B then the curve represents early leakers or infant mortality. When B = 3.34, then a normal distribution results. When B>3.34, then a skewed distribution representing a wear-out mode is the result.

The characteristic life (θ) is defined to be that point in time where 63.7% of the seals tested have failed. This parameter controls the spread of the failure rate distribution curve (Fig. 7.22). Small values of B result in a tight

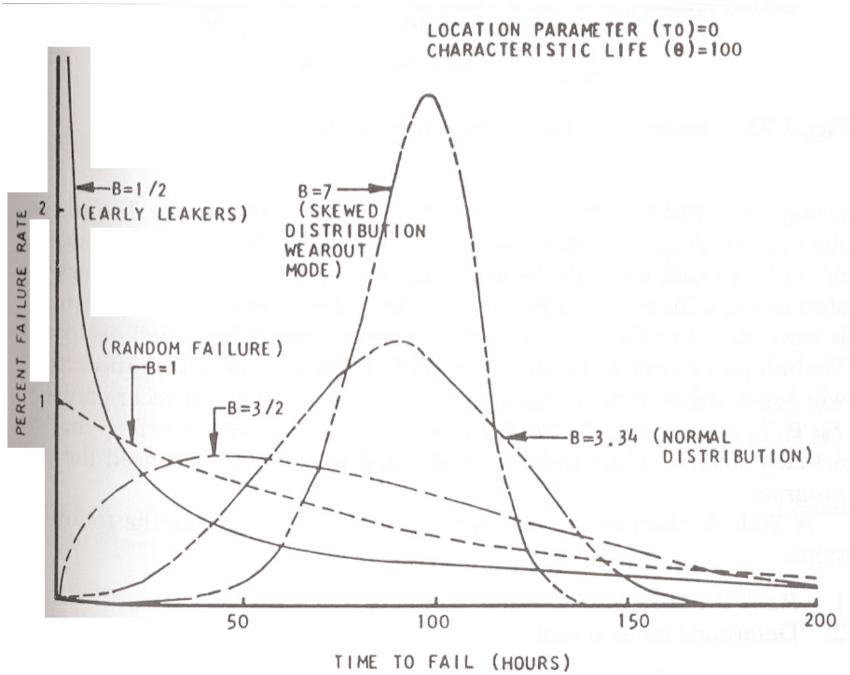


Fig. 7.21 The effect of shape parameter (B) on failure rate distribution.

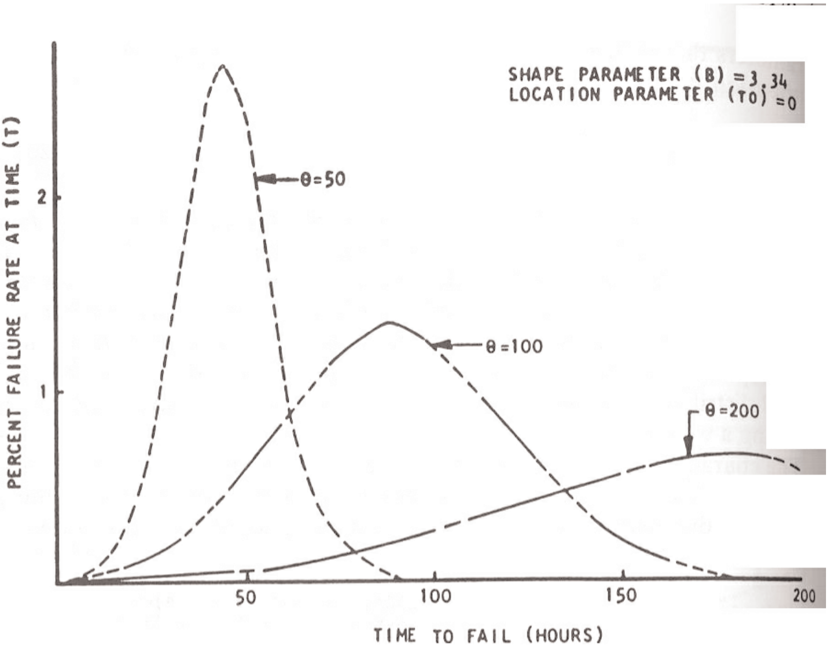


Fig. 7.22 The effect of characteristic life (θ) on failure rate distribution.

curve, and most failures occur within a short time frame. As θ increases, the curve becomes broader, but the type of failure mode (infant, random, normal, skewed) does not change. The location parameter (T_0) defines the start of the failure distribution curve (Fig. 7.23). If the location parameter is zero, then the Weibull curve will be linear when it is plotted on special Weibull paper with adjusted scales, and the failure rate distribution curve will begin at time zero. If T0 is positive, then no failures will occur until T = T0. If T0 is negative, the implication is that failures began before the test actually started, which indicates that scrap parts have influenced the test program.

A Weibull chart can be developed and plotted by taking the following steps:

1. Rank the life data.
2. Determine median rank:

$$F_{T50} = \frac{j - 0.3}{N_T + 0.4} \tag{7.21}$$

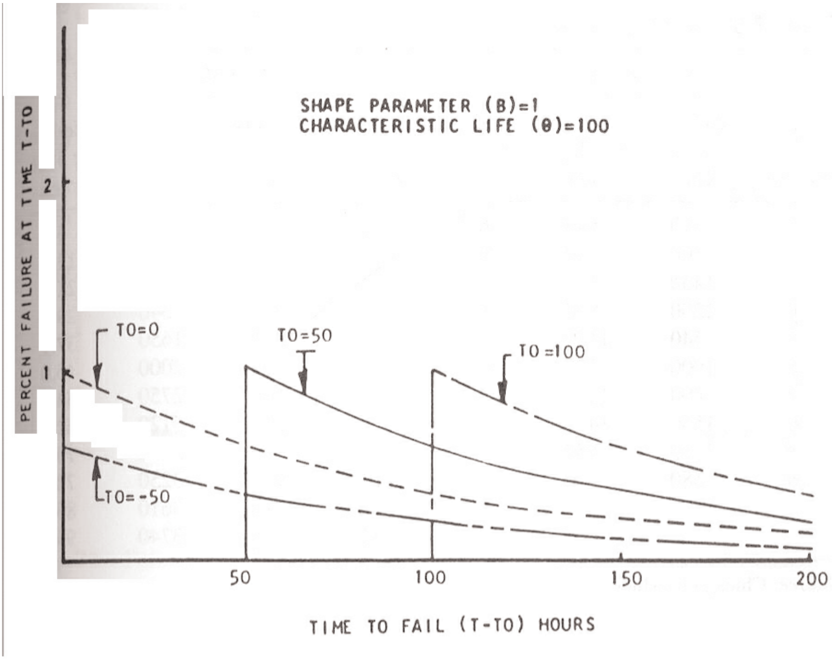


Fig. 7.23 The effect of location parameter (T_0) on failure rate distribution.

Where F_{T50} is the cumulative percent failure for median rank and j is the rank of the test.

3. Plot data on Weibull paper.
4. Determine location parameter (T_0),

$$T_0 = T_2 - \frac{(T_3 - T_1)(T_2 - T_1)}{(T_3 - T_1) - (T_2 - T_1)} \tag{7.22}$$

Where T1, T2, and T3 are the time for first failure, midpoint failure from Weibull curve, and last failure, respectively.

5. Adjust the data.
6. Determine the Weibull slope B.
7. Determine the characteristic life θ.
8. Determine the number and types of failure modes.
9. Establish confidence limits.

Table 7.9 Seal Failure Data

	Seal type A		Seal type B				
.	Failure	Median	Failure				Median
	time	Ranked	or 50%		time	Ranked	or 50%
Seal no	(hr)	data	rank	Seal no.	(hr)	data	rank
1	610	540	6.7	1	540	175	5.6
2	1100	580	16.3	2	3740	200	13.7
3	1450	610	25.9	3	175	300	21.8
4	1200	800	35.6	4	1650	540	29.8
5	540	1100	45.2	5	2000	1650	37.9
6	1900	1150	54.8	6	200	2000	46.0
7	800	1200	64.4	7	3610	2750	54.0
8	1550	1450	74.1	8	3200	3120	62.1
9	1150	1550	83.7	9	300	3200	70.2
10	580	1900	93.3	10	3250	3250	78.2
				11	3120	3610	86.3
				12	2750	3740	94.4

Source: Chicago Rawhide.

Consider the following example. Failure data for 10 seals from population A and 12 seals from population B appear in Table 7.9. The data are first ranked in order of time of leakage. The cumulative percent failure is then determined for each of the failure points. The median or 50% rank is the most likely value for the cumulative percent failure, and it is the one used to make the Weibull plot. The medium point has a 50% chance of this time being larger or smaller than the actual failure point for the population. Other curves appearing in tables can give the various rank curves. If a 95% rank curve is plotted, then the actual failure point has a 95% probability of being larger than the point that appears on the 95% rank curve. The median rank is usually calculated with Equation (7.21). The data are then plotted using the specialized Weibull paper (Fig. 7.24). The plot is nonlinear, which suggests that several failure modes may be present, which is typical for production seals. The location parameter T0 is then estimated using Equation (7.22). An alternative approach is to estimate the intercept on a linear cumulative percent failure curve (Fig. 7.25). The latter approach is recommended if the Weibull curve is nonlinear. The location parameter T_0 is estimated to be 540 hr for seal type A and 140 hr for seal type B. After the data are adjusted, the lower portion of the Weibull curve becomes linear. This is illustrated in Fig. 7.24 for seal type B. The slope of the curve becomes greater above 2000 hr. This indicates that there are at

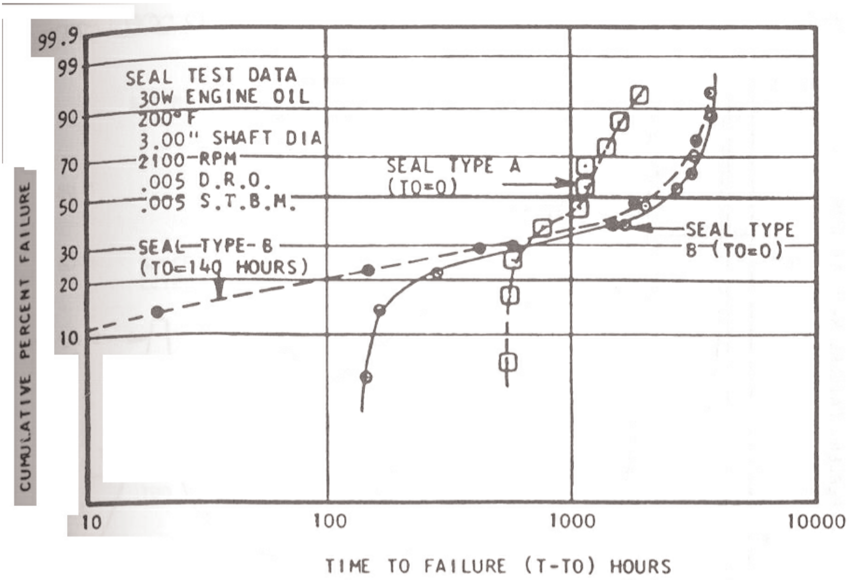


Fig. 7.24 Failure data for two seal types (Weibull plot). (Courtesy of Chicago Rawhide.)

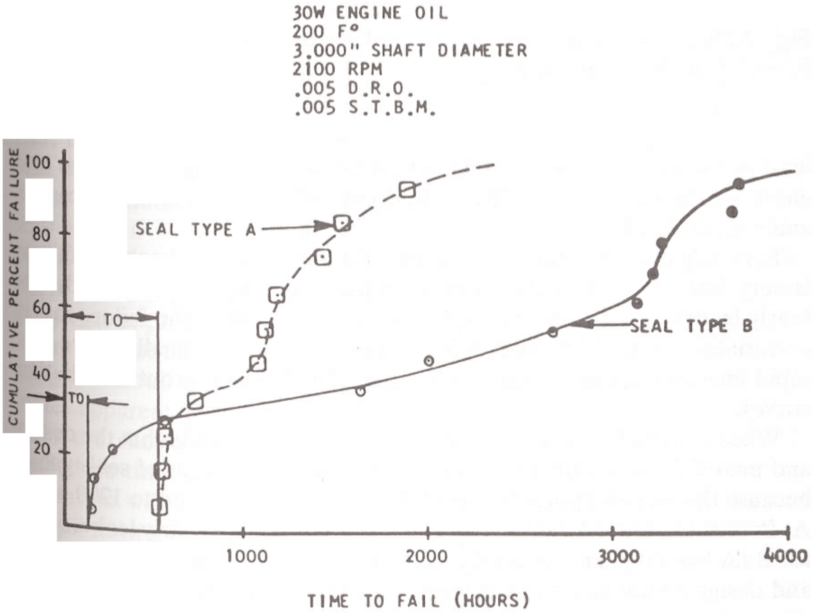


Fig. 7.25 Estimation of T0 using linear plots. (Courtesy of Chicago Rawhide.)

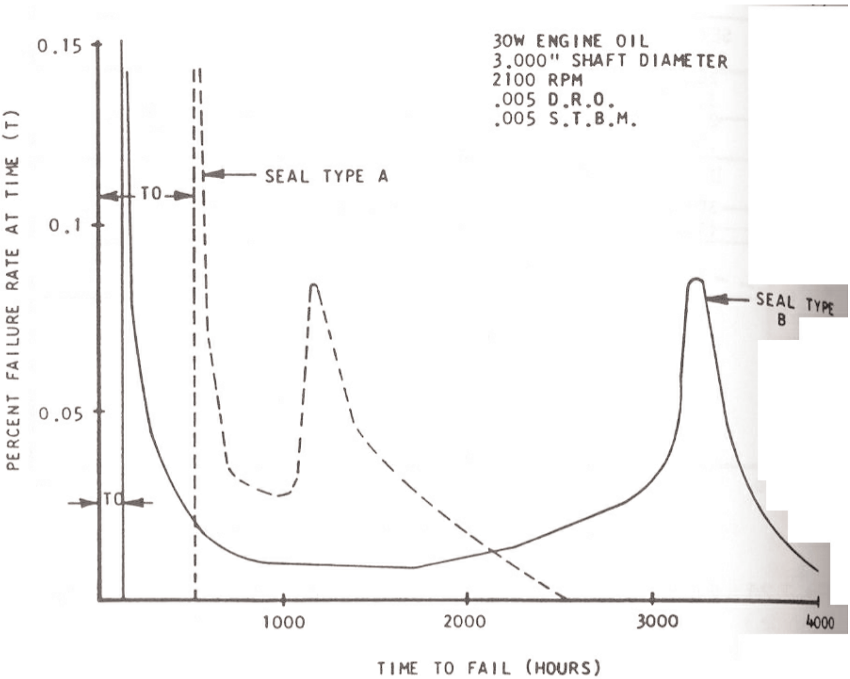


Fig. 7.26 Failure rate versus time to fail for two different seal populations. (Courtesy of Chicago Rawhide.)

least two modes of failure present, which can be highlighted if a failure rate chart is prepared from the Weibull curves. This chart (Fig. 7.26) can be analyzed in detail.

For each seal type, there is a period of time when the chance of failure is very low (T_0). The failure rate then leaps to a high value and decays (early leaker distribution curve). There is a period when the failure rate is essentially constant (random failure distribution), and finally there is a rapid increase or peak in the failure rate (skewed or wearout distribution curve).

When comparing the two seal types, one would conclude that the design and material combination of seal type B is superior to that of seal type A because the wearout point occurs at 3100 hr for B compared to 1200 hr for A. Processing procedures for A give a lower incidence of early leakers than for B (A has a higher T_0 than B). Seal type B thus has the superior material and design combination but requires process improvements to reduce the percentage of early leakers.

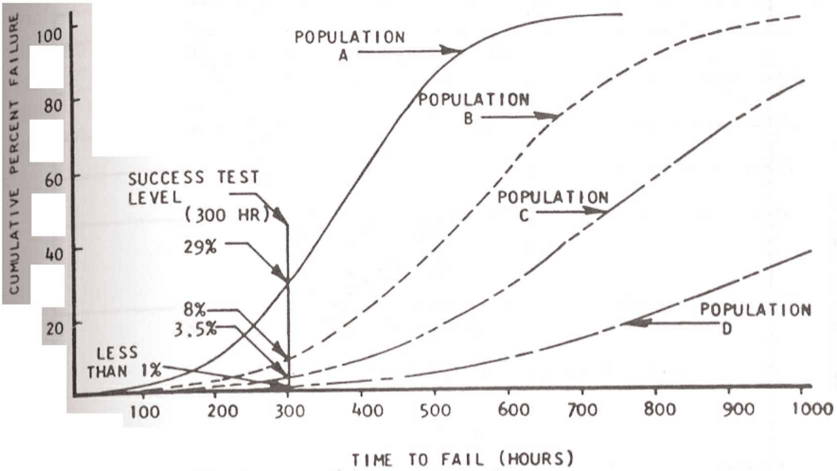


Fig. 7.27 Cumulative percent failure for four seal populations. (Courtesy of Chicago Rawhide.)

7.5.4. Success Versus Weibull Testing

Consider a success test that requires all seals to pass a 300 hr approval test without leakage. The seals that are tested are selected from four different large populations that have cumulative percent failure rates shown in Fig. 7.27. Even if no failures occur during success testing, the probability that 99% of the population will pass the 300 hr test is small.

For population A (with 29% failures at 300 hr), it is likely that three seals will fail before the 300 hr mark; thus the chance of approving the design based on success testing is small. The chance of 10 seals passing the 300 hr mark is good for seals from populations B and C. It is thus quite likely that approval could be granted to these seal types even though the failure level of the population at 300 hr is 3.5-8%. By testing to failure and using a Weibull plot (Fig. 7.28), the shape of the distribution curve becomes apparent and confidence in the results improves. The 70 and 90% ranks are shown for 10 seals tested from population D. There is 90% confidence that less than 3% and a 70% confidence that only 1% of the population will fail before 300 hr. The comparisons between success and Weibull testing for population D appears in Table 7.10. Ten seals tested to failure and analyzed using Weibull statistics require 13,130 hr and will generate a 70% confidence level that 99% of population D will pass the 300 hr test. To achieve the same 70% confidence level using success testing, 125

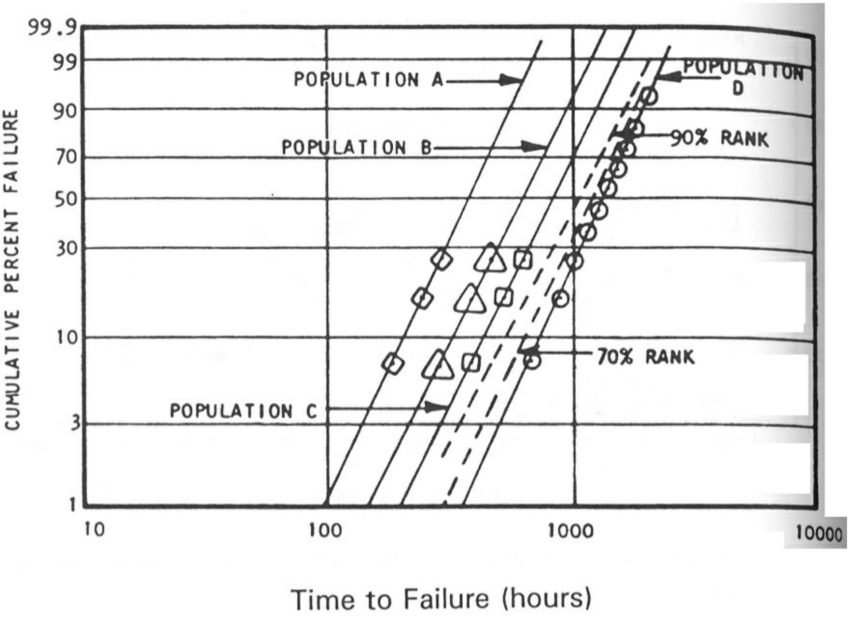


Fig. 7.28 Weibull plot for four seal populations. (Courtesy of Chicago Rawhide.)

Table 7.10 Comparison of 300 hr Success Tests and Weibull Testing to Failure-Seal Population D (99% Reliability Required)

Test type	Number of seals tested	Number of failures	Percent confidence that population D leakage is less than 1% for the first 300 hr	Required test hours
Success	8	0	7	2,400
	12	0	11	3,600
	70	0	50	21,000
	125	0	70	37,500
	8	1	0.5	2,400
	12	1	1	3,600
	70	1	15	21,000
	250	1	70	75,000
	270	2	50	81,000
Weibull	10	10	70	13,130
	10	4	70	10,490

Source: Ref. 73 and Chicago Rawhide.

seals with no leakers must be tested, which consumes 37,500 test hours. If one leaker occurs, then an additional 125 seals must be tested without leakage, which requires a total of 75,000 test hours. It is obvious that this quantity of testing is prohibitive and Weibull analysis is preferred to achieve high confidence levels.

8

Effect of Operating Parameters on Seal Performance

8.1 BASELINE TEST CONDITIONS

A series of tests were conducted to determine what effect varying the operating parameters has upon shaft seal performance [82, 83]. The underlip temperature, torque, pump rate, and seal life were determined using the techniques described in Chapter 7. The results are to be considered trends and are not intended to be representative of seal life operating under field conditions. The parameters varied during the tests were shaft speed, sump temperature, sump pressure, dynamic runout, shaft-to-bore misalignment, seal cock, fluid viscosity, and sump level. The effect of a specific variable was determined by changing that variable while the others were kept constant. Any other deviation from the base conditions is noted. The elastomeric material used in the study was nitrile (NBR). Seals for the test were manufactured from the same batch of NBR material and were molded in a single-cavity mold using identical process conditions during a single manufacturing run. The values of the parameters that define the base test conditions are given in Table 8.1.

Table 8.1 Baseline Conditions for Life Tests

Shaft size	76.2 mm (3.000 in.)
Bore size	101.6 mm (4.000 in.)
Seal width	12.7 mm (0.500 in.)
Seal elastomeric material	Nitrile (NBR)
Shaft speed	2165 rpm, 8.63 m/sec (1700 ft/min)
Sump temperature	93°C (200°F)
Sump pressure	Zero
Fluid level in sump	Centerline
Test fluid	SAE 30 engine oil
Dynamic runout (DRO)	0.13 mm (0.005 in.)
Shaft-to-bore misalignment (STBM)	0.13 mm (0.005 in.)
Seal cock	Zero
Test cycle: 20 hr at test speed and temperature, 4 hr at zero speed, ambient temperature	

Source: Chicago Rawhide.

8.2 EFFECT OF SUMP TEMPERATURE ON SEAL PERFORMANCE

Measurements of seal lip temperature, torque, and pump rate were made simultaneously using the techniques described in Chapter 7. The seal lip temperature increases as shaft speed and sump temperature increase (Fig. 8.1).

When the sump temperature is increased at a given speed, the fluid viscosity decreases, which results in lower torque and lower power consumption. Increasing shaft speed at a constant sump temperature increases frictional drag, which increases torque and power consumption (Figs. 8.2 and 8.3). The seal's inherent pumping ability increases as speed and sump temperature increase (Fig. 8.4). The net result of an increase in sump temperature is a reduction of seal life (Table 8.2) [55, 82]. The increase in temperature causes an increase in the hardening rate of the nitrile material, which causes a loss of flexibility and followability of shaft dynamics. This hardening is a temperature-dependent chemical reaction that roughly follows the Arrhenius law,

$$L = \alpha \exp\left(\frac{E}{RT_K}\right) \tag{8.1}$$

where L is seal life (hr), α is the life constant for a given material (hr), E is the energy of activation, R is the gas constant, and TK is sump temperature (K).

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Lubricant	SAE 30
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 inch)
Seal cock	Zero
Pressure	Zero
Sump level	Full

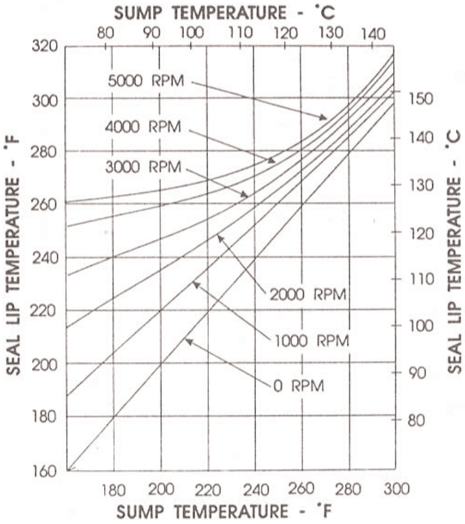


Fig. 8.1 Underlip temperature versus sump temperature for various shaft speeds. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR
Shaft size	76.2mm (3.000 inch
Lubricant	SAE 3
DRO	0.13mm (0.005 inch
STBM	0.13mm (0.005 inch
Seal cock	Zer
Pressure	Zer
Sump level	Fu

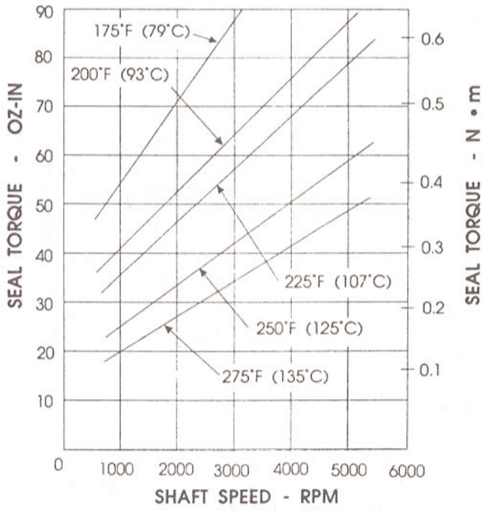


Fig. 8.2 Seal torque versus shaft speed for various sump temperatures. Parameters are the same as for Fig. 8.1. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Lubricant	SAE 30
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 inch)
Seal cock	Zero
Pressure	Zero
Sump level	Full

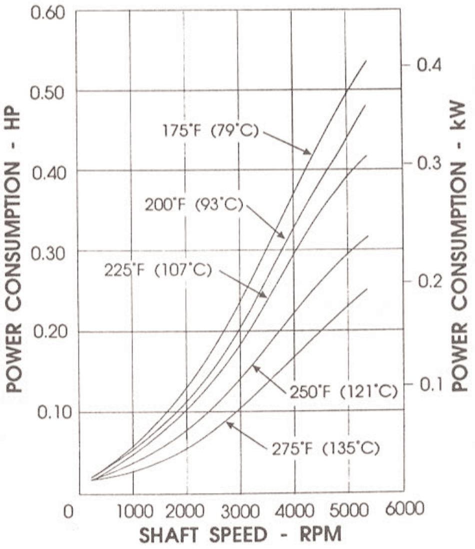


Fig. 8.3 Seal power consumption versus shaft speed for various sump temperatures. Parameters are the same as for Fig. 8.1. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Lubricant	SAE 30
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 INCH)
Seal cock	Zero
Pressure	Zero
Sump level	Full

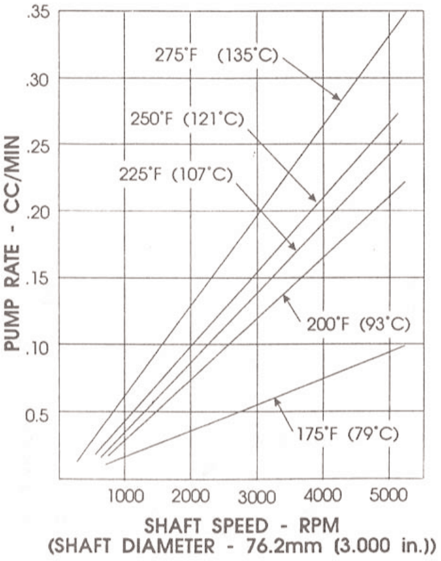


Fig. 8.4 Seal pump rate versus shaft speed for various sump temperatures. Parameters are the same as for Fig. 8.1. (Courtesy of Chicago Rawhide.)

Seal life at a given temperature can be predicted for the nitrile used in the test with the equation

$$L_{\mathrm{NBR}} = 8.54 \times 10^{-7} \exp\left(\frac{7905}{T_{\mathrm{K}}}\right) \tag{8.2}$$

Life of the polyacrylate (PA) material can be estimated with the equation.

$$L_{\mathrm{PA}} = 113.59 \times 10^{-7} \exp\left(\frac{7905}{T_{\mathrm{K}}}\right) \tag{8.3}$$

A general rule of thumb is that seal life is doubled for a given elastomer for every 14°C (25°F) decrease in sump temperature.

Table 8.2 Seal Life Versus Sump Temperature

Shaft size	76.2 mm (3.000 in.)
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Shaft speed	2165 rpm
Cycle	20 hr on, 4 off
Sump level	Centerline

Sump temp. (°C)	Seal life (hr)		
	Nitrile (GL-5)	Nitrile (SAE 30)	Polyacrylate (PA) (SAE 30)
79	--	4875	--
93	1950	2070	--
107	900	950	--
121	500	575	5750
135	--	250	2950
148	--	--	1750
163	--	--	850

Source: Chicago Rawhide.

The activation energy is the same for both polymers. The frequency factor for the PA materials is about 13 times larger than for the nitrile material. This means that the PA material will require 10--15 times as long to harden and leak than the nitrile material at the same sump temperature. Other premium elastomeric materials such as silicone, fluoroelastomer, or hydrogenated nitriles will last even longer.

As shaft speed increases at a constant temperature, the underlip temperature will increase and seal life will be shorter (Table 8.3). Life decreases by a factor of 2 as the shaft speed increases by a factor of 10 (e.g., from 500 to 5000 rpm).

Radial lip seals are friction devices, and the primary sealing lip will wear. The width of the wear track depends upon shaft speed (Table 8.3). Under the given test conditions, the average wear track width is about 0.020 in. at 500 rpm. At 5000 rpm, the average wear track width has doubled even though the average life is one-half the life obtained at 500 rpm. It may be necessary to choose premium wear-resistant materials if high shaft speeds are expected in the application.

Table 8.3 Seal Life Versus Shaft Speed

Material	Nitrile (NBR)
Shaft size	76.2 mm (3.000 in.)
Lubricant	SAE 30
DRO	0.13 mm (0.005 in.)
STEM	0.13 mm (0.005 in.)
Seal cock	Zero
Seal pressure	Zero
Sump temperature	121°C (250°F)
Cycle	20 hr on, 4 off
Sump level	Full

		Seal wear track width (mm)		
Shaft speed (rpm)	Seal life (hr)	Lower limit	Mean	Upper limit
500	780	0.44	0.50	0.60
1000	650	0.46	0.53	0.62
2000	510	0.48	0.57	0.66
3000	420	0.51	0.64	0.76
4000	380	0.64	0.81	0.94
5000	350	0.76	1.02	1.14

Source: Chicago Rawhide.

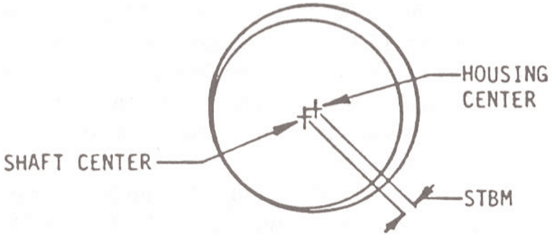


Fig. 8.5 Shaft-to-bore misalignment (STBM). (Courtesy of Chicago Rawhide.)

8.3 EFFECT OF ECCENTRICITY ON SEAL PERFORMANCE

The shaft-to-bore misalignment (STBM) is the distance by which the center of the shaft is offset from the center of the bore (Fig. 8.5). STBM is measured by attaching the base of a dial indicator to the shaft and allowing the ball of the indicator to ride on the inside of the bore. The shaft is slowly rotated, and the difference between the maximum and minimum readings observed when a complete revolution is made is recorded. This reading is STBM or offset. STBM is the result of normal machining tolerances and assembly stackup inaccuracies. Excessive STBM will result in light interference and light wear on one side of the seal with heavy interference and wear at a point 180° away from the light spot. For good performance, STBM should be 0.25 mm or less.

Dynamic runout (DRO) is caused by misalignment, shaft flexing, shaft vibrations due to imbalance, shaft lobing, and other manufacturing inaccuracies. It is the amount in millimeters or inches by which the shaft surface does not rotate about the true shaft center. DRO will cause a local point on the seal lip to move in a radial direction as the shaft rotates. The seal lip section design and material must provide flexibility to follow the radial motion of the shaft without allowing gaps to form. When flexibility is lost, leakage will occur when gaps form between the rotating shaft and the seal lip. DRO is measured by attaching the base of a dial indicator to the face of the housing and allowing the ball to ride on the surface of the shaft when the shaft is slowly rotated. The total movement of the ball is recorded. This method cannot be used in actual applications when the shaft is rotating at high speeds. Electronic devices are available that can measure DRO at high speeds without contacting the shaft surface. In general, to ensure good performance, the dynamic runout should not exceed 0.25 mm (0.010 in.) total indicated reading (TIR).

The amount of eccentricity that a seal can withstand depends on shaft speed [3--5, 120, 143, 144.] Typical spring-loaded seals will operate satisfactorily if the total eccentricity (STBM and DRO) does not exceed the maxima shown in Table 8.4 for the range of shaft speed. Special seal designs with convoluted sealing lips may be required if the eccentricity exceeds these limits.

Changing STBM and DRO from 0.0 to 0.5 mm (0.020 in.) had no noticeable effects on underlip temperature, seal torque, power consumption, and pumping ability. Seal life was unaffected by changes in STBM over the normal range (0.0--0.5 mm) (0.0--0.020 in.). Seal life changed dramatically when the DRO was varied (Table 8.5). The mean life of nitrile seals was 550 hr with zero dynamic runout at 250°F and 2165 rpm (8.63 m/sec or 1700 ft/min). Seals leaked instantly when the dynamic runout exceeded 0.6mm (0.024 in.).

8.4 EFFECT OF SEAL COCKING ON SEAL PERFORMANCE

After installation, the back face of the seal may not be perpendicular to the centerline axis of the shaft. This condition is known as angular misalignment or seal cock. This angular misalignment results in a one-cycle wave

Table 8.4 Maximum Total Eccentricity (STBM and DRO) Versus Shaft Speed

Maximum shaft speed		Maximum total eccentricity	
ft/min	m/sec	in.	mm
75	0.38	0.025	0.635
150	0.76	0.020	0.508
350	1.78	0.018	0.457
700	3.55	0.015	0.381
1000	5.08	0.013	0.330
1500	7.61	0.010	0.254
1800	9.14	0.009	0.228
2000	10.16	0.008	0.203
3000	15.23	0.007	0.178

Source: Chicago Rawhide.

Table 8.5 Seal Life vs. Dynamic Runout

Material	Nitrile (NBR)
Shaft size	76.2 mm (3.000 in.)
Lubricant	SAE 30
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Shaft speed	2165 rpm
Shaft temperature	121°C (250°F)
Cycle	20 hr on, 4 off
Sump level	Centerline

Dynamic runout (mm)	Mean seal life (hr)
0.0	550
0.1	460
0.2	365
0.3	275
0.4	180
0.5	90
0.6	0.0

Source: Chicago Rawhide.

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Sump temperature	93°C (200°F)
Lubricant	SAE 30
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 inch)
Pressure	Zero
Sump level	Full

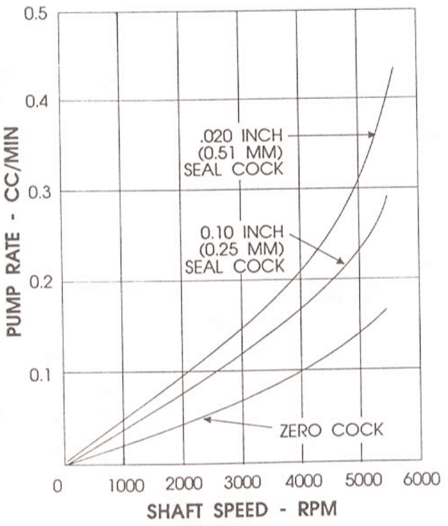


Fig. 8.6 Seal pump rate versus shaft speed for various values of seal angular misalignment. (Courtesy of Chicago Rawhide.)

seal that can increase seal pumping as the amplitude of the cocking increases (Fig. 8.6). The underlip temperature, seal torque, and power consumption also increase (Figs. 8.7--8.9). Seal life was unaffected by cocking up to 0.5 mm (0.020 in.). Excessive seal cocking beyond 0.5 mm can restrict the seal lip, creating premature damage that results in early leakage. It is essential to use proper installation techniques to minimize seal cocking and avoid premature leakage due to the damage that may occur.

8.5 EFFECT OF LUBRICANTS ON SEAL PERFORMANCE

The type of media to be sealed must be known in order to select the proper sealing element material. The selection should be based on the operating temperature and chemical compatibility.

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Sump temperature	93°C (200°F)
Lubricant	SAE 30
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 inch)
Pressure	Zero
Sump level	Full

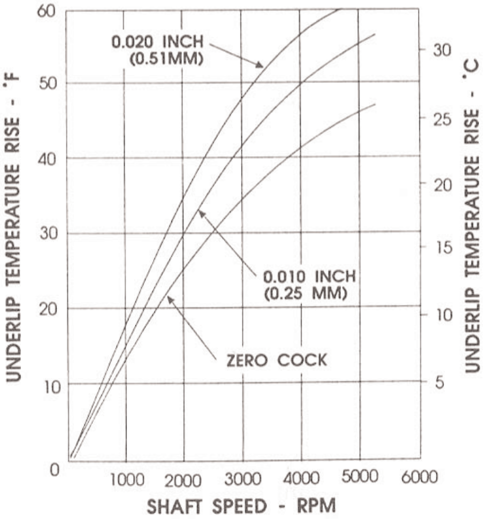


Fig. 8.7 Underlip temperature (°C) versus shaft speed for various values of seal angular misalignment. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Sump temperature	93°C (200°F)
Lubricant	SAE 30
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 inch)
Pressure	Zero
Sump level	Full

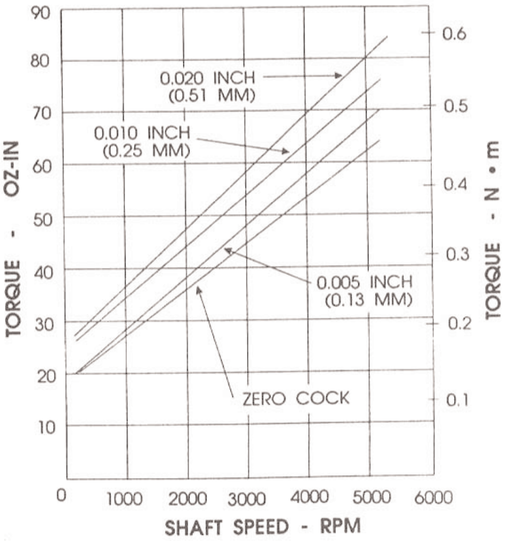


Fig. 8.8 Seal torque versus shaft speed for various values of seal angular misalignment. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Sump temperature	93°C (200°F)
Lubricant	SAE 30
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 inch)
Pressure	Zero
Sum level	Full

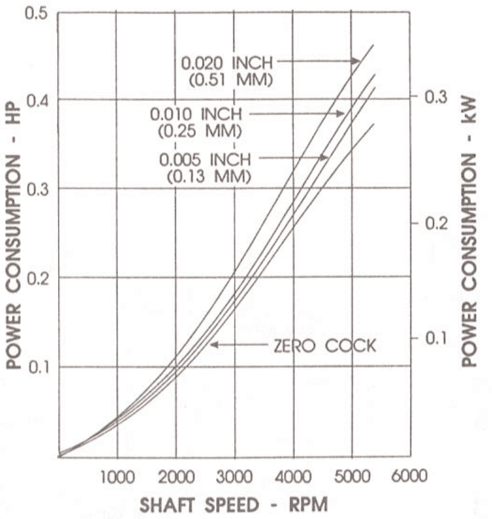
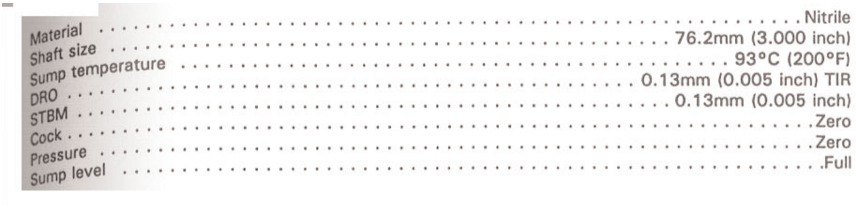


Fig. 8.9 Power consumption versus shaft speed for various values of seal angular misalignment. (Courtesy of Chicago Rawhide.)

Modern lubricants employ many chemicals to improve their performance. Unfortunately, additives that improve the lube may be incompatible with the seal material. For example, disulfide additives give lubricants antiwear properties, but they also cure or harden the sealing element. Many of the additives in EP (extreme pressure) lubricants increase their chemical activity as they heat up, and they affect the rubber far more at 110°C (230°F) than at 82°C (180°F). Some materials revert and fall apart. When the sealing member softens excessively with use, or whenever there is not sufficient heating to explain the hardening observed, most likely the lubricant and seal material are chemically incompatible. The remedy lies in proper seal and fluid selection.

Different applications may require fluids with different viscosities. Viscosity decreases as temperature increases, and this will affect frictional torque, power loss, pump rate, and seal life. Fluids with high viscosities



Material	Nitrile
Shaft size	76.2mm (3.000 inch)
Sump temperature	93°C (200°F)
DRO	0.13mm (0.005 inch) TIR
STBM	0.13mm (0.005 inch)
Cock	Zero
Pressure	Zero
Sump level	Full

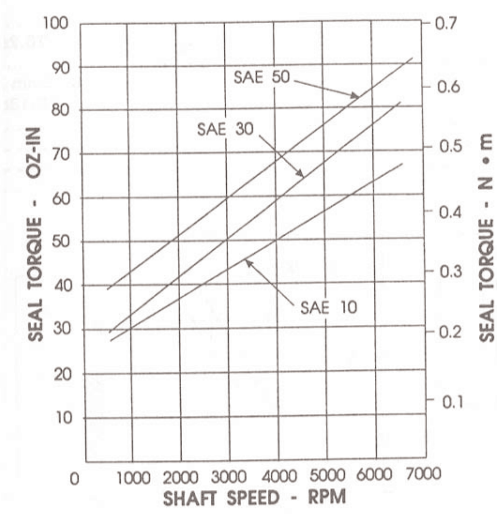


Fig. 8.10 Seal torque versus shaft speed for various fluid viscosities. (Courtesy of Chicago Rawhide.)

generate more friction than low-viscosity fluids, which results in higher torque (Fig. 8.10) and power loss (Fig. 8.11). The pump rates of low-viscosity fluids are slightly higher than the pump rates of high-viscosity fluids (Fig. 8.12).

Fluids can also affect other parts of the sealing system. A test was run to determine the effect of fluid types on seal life and shaft surface. The fluids evaluated were SAE 30 engine oil and a GL-5 EP gear lube of similar viscosity. The seal life was almost equivalent (Table 8.2).

The shaft wear rate was dramatically different (Table 8.6). At 93°C 200°F), the shaft wear rate of the GL-5 lube was twice that of the engine oil. When the temperature was increased to 107°C (225°F), the shaft wear rate in GL-5 was 6.5 times that of the engine oil. The shaft wear in GL-5 is caused by the additive package, which creates a chemical etching process.

Material	Nitrile
Shaft size	76.2mm (3.000 inch)
Sump temperature	93°C (200°F)
DRO	0.13mm (0.005 inch) TIR
STBM	0.13mm (0.005 inch)
Cock	Zero
Pressure	Zero
Sump level	Full

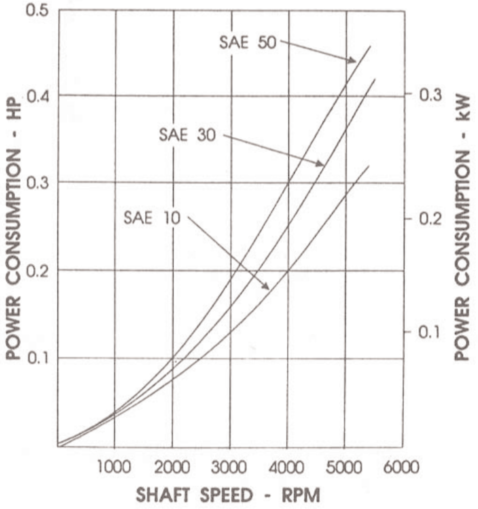


Fig. 8.11 Power consumption versus shaft speed for various fluid viscosities. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Shaft speed	2165 RPM
DRO	0.13mm (0.005 inch)
STBM	0.13mm (0.005 inch)
Cock	Zero
Pressure	Zero
Sump level	Full

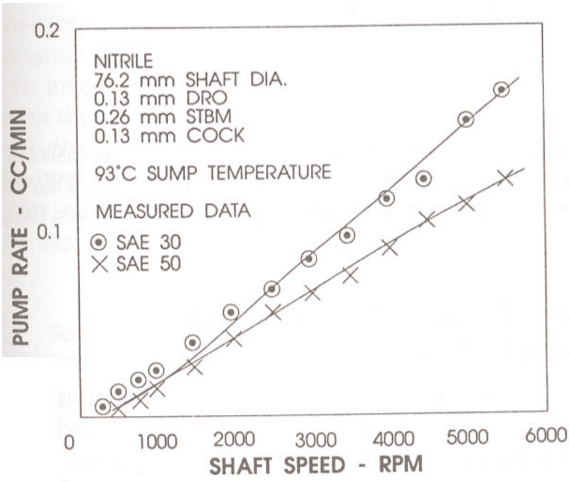


Fig. 8.12 Pump rate versus fluid viscosity. (Courtesy of Chicago Rawhide.)

Table 8.6 Shaft Wear Rate (μin./hr) vs. Fluid Type

Material	Nitrile (NBR)
Shaft size	76.2 mm (3.00 in.)
Shaft speed	2165 rpm
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Cock	Zero
Pressure	Zero
	Cycle 20 hr on, 4 off
Sump level	Centerline

93°C (200°F)		107°C (225°F)	
SAE 30	GL-5	SAE 30	GL-5
15	30	17	113

Source: Chicago Rawhide.

This additive package is not present in the engine oil. Excessively worn shafts must be replaced or refinished when seals are replaced. This expense can be eliminated by using chrome-plated shafts or wear sleeves that resist chemical attack.

8.6 EFFECT OF LUBRICANT FILL LEVEL ON SEAL PERFORMANCE

Lip seals should not be run without lubrication for any prolonged period of time [57]. As sump temperature or speed increases, the lubricating film thins and may break down. In many applications, lubrication in the seal area is sparse or nonexistent for a brief period of time before the oil reaches the seal. Under these circumstances, direct contact between the lip and shaft increases, resulting in high wear or stick-slip, which may destroy the seal. Stick-slip is a friction phenomenon that manifests as jerky motion between two surfaces due to alternate gripping and slipping of contact areas. This action produces a surface wave oscillation that can literally tear a seal apart or at best allow leakage. Different seal materials can behave quite differently when run dry. A test was run with the same design but with three different materials to illustrate the phenomenon. Seal A ran smooth for 5 min, but went into stick-slip after climbing to an underlip temperature of 475°F (246°C). Seal B ran for the duration of the test without stick-slip and never reached an underlip temperature greater than 275°F (135°C). Seal C climbed to an excessively high temperature of 500°F (20°C) and went into such violent stick-slip that it destroyed itself within 2 min.

Seals compounded without dry lubricants and seals made of low-modulus materials generally tend to stick-slip worse than others. Some seals that run well dry may develop stick-slip when sparsely lubricated. When stick-slip develops, the frictional torque will increase dramatically, which causes the lip temperature to escalate rapidly. Failure can result quickly. Presoaking silicone seals, which absorb oil, or the use of nitriles or Teflon materials that wear well will alleviate this type of problem in sparse lubrication.

Changing the sump fill level affects seal performance and life [164, 229]. As sump level is decreased, the seal life decreases (Table 8.7). The reduction in life results because the lower quantity of oil cannot remove the frictional heat as effectively as a full head of oil. As a result, the underlip temperature increases (Fig. 8.13) even though the frictional torque and power consumption decrease (Figs. 8.14 and 8.15) as the fill level is reduced. Vassmer [229] conducted a mathematical study using finite-element analysis to investigate why lip temperature increases as sump level decreases even though frictional torque decreases. This study revealed that at high speeds the oil in a full sump removed about 85% of the heat generated. When the sump level drops to quarter-shaft, the oil removes less than 70% of the heat generated. As the oil level decreases, the convection cooling of the seal decreases. Even though less heat is generated at low

Table 8.7 Seal Life vs. Sump Fill Level

Material	Nitrile (NBR)
Shaft size	76.2 mm (3.000 in.)
Shaft speed	5000 rpm, 20 m/sec (3927 ft/min)
Lubricant	SAE 30
Sump temperature	121°C (250°F)
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Cycle	20 hr on/4 hr off

Sump fill level	Seal life (hr)
Full	1745
Half-shaft	633
Quarter-shaft	475

Source: Chicago Rawhide.

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Lubricant	SAE 30
Sump temperature	93°C (200°F)
Total eccentricity	0.025mm (0.002 inch)
Seal cock	Zero
Pressure	Zero

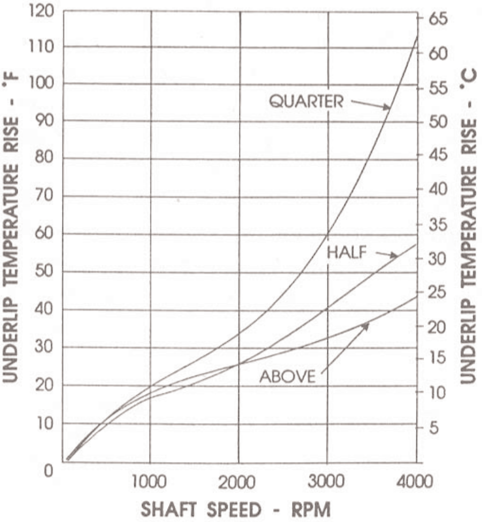


Fig. 8.13 Temperature rise versus shaft speed for various sump fill levels. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Lubricant	SAE 30
Sump temperature	93°C (200°F)
Total eccentricity	0.025mm (0.001 inch)
Seal cock	Zero
Pressure	Zero

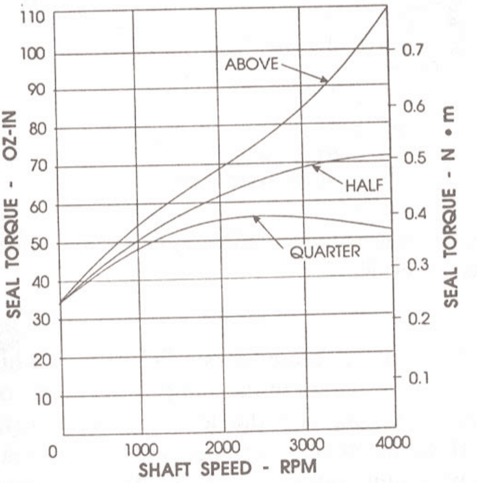


Fig. 8.14 Seal torque versus shaft speed for various sump fill levels. (Courtesy of Chicago Rawhide.)

Material	Nitrile (NBR)
Shaft size	76.2mm (3.000 inch)
Lubricant	SAE 30
Sump temperature	93°C (200°F)
Total eccentricity	0.025mm (0.001 inch)
Seal cock	Zero
Pressure	Zero

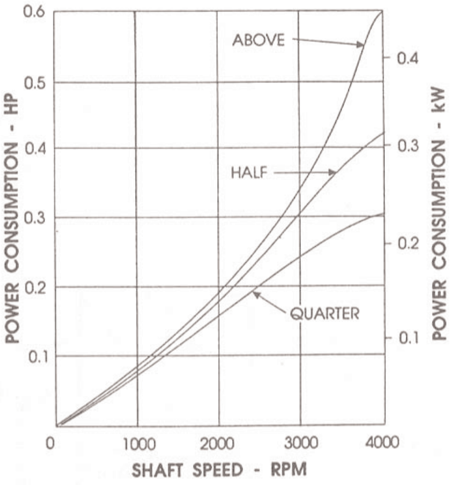


Fig. 8.15 Seal power consumption versus shaft speed for various sump fill levels. (Courtesy of Chicago Rawhide.)

sump fill levels, it cannot be removed as efficiently, and high lip temperatures result in lower life. The oil in the sump is a critical component in the cooling system for the seal, and the level should never be allowed to become too low. If the machine is designed so the seal area receives sparse lubrication, then premium materials such as PTFE may be required.

8.7 EFFECT OF PRESSURE ON SEAL PERFORMANCE

Standard seal designs and materials are not suitable for high-pressure, high-speed applications. Excessive pressure forces the entire air-side section of the seal down onto the shaft, which results in excessive wear (Fig. 8.16). In some cases, the seal tip no longer touches the shaft and the load is carried entirely by the air-side seal surface. This condition is known as bell mouthing and is detected when the seal lip has little or no wear and the air-side seal surface shows evidence of heavy wear. Some applications may inadvertently generate high pressures if air vents become clogged with dirt or paint. Extreme pressure buildup can force the seal out of the bore, destroy the bond between the rubber and the metal case, wear out the seal lip, and severely groove the shaft. Early leakage will result, and in some cases the leakage will be catastrophic. The optimum life of conventional seal designs is at “zero” pressure, and this life will decrease as the pressure and shaft surface speed increase.

The maximum operating pressure limits for conventional seal designs as a function of shaft surface speed are given in Table 8.8. It must be emphasized that the maximum operating pressure for standard seal designs is only 48.3 kPa (7 psi). Life tests dramatically illustrate the effect of pressure on standard seal life (Table 8.9). The life of a conventional seal design with a nitrile elastomeric lip is over 2000 hr with zero pressure, and the average seal lip wear is 0.63 mm (0.025 in.). When the sump pressure is increased to 103.4 kPa (15 psi), the seal lip wear increases to 2.92 mm (0.115 in.) and the seal life drops to 179 hr. The effect of sump level is also demonstrated. The average life at 103.4 kPa (15 psi) increases from 179 to 497 hr when the lubricant level increases from shaft centerline to full.

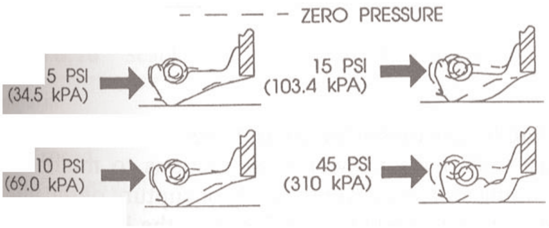


Fig. 8.16 Pressure distortion of standard seal designs. (Courtesy of Chicago Rawhide.)

Table 8.8 Maximum Operating Pressure Limits vs. Speed for Standard Seal Designs

Shaft speed		Maximum pressure	
ft/min	m/sec	psi	kPa
0--1000	5.076	7	48.3
1001--2000	5.081--10.152	5	34.5
2001 and higher	10.157 and higher	3	20.7

Source: Chicago Rawhide.

Table 8.9 Seal Life vs. Operating Pressure for Standard Seal Designs

Material	Nitrile (NBR)
Shaft size	76.2 mm (3.000 in.)
Shaft speed	2165 rpm (8.629 m/sec, 1700 fpm)
Sump temperature	93°C (200°F)
Lubricant	SAE 30
STBM	0.13 mm (0.005 in.)
DRO	0.13 mm (0.005 in.)
Cock	Zero
Sump level	Centerline (except where indicated)
Cycle	20 hr on, 4 hr off

Sump Pressure			Seal wear width (mean value)	
kPa	psi	Seal life	mm	in.
0	0	2070	0.63	0.025
34.5	5	1332	1.14	0.045
69.0	10	1217	2.41	0.095
103.4	15	179	2.92	0.115
103.4	15[a]	477	--	--

[a] Full sump.

Source: Chicago Rawhide.

In applications involving higher pressures, special pressure seal designs should be used. Pressure seals have heavier cross sections to minimize distortion and greater bonding area to prevent tearing or rupture of the lip. These seals can accommodate up to 690 kPa (100 psi) at the lower speed range. For extreme pressures of up to 10,350 kPa (1500 psi) at slow speeds, antiextrusion rings may be incorporated behind the primary lip as a backup support. When using pressure seals, a retainer washer, such as a snap ring or bolt-on flange, must be inserted behind the seal and mounted to the housing bore to prevent the seal from being forced out by pressure.

8.8 EFFECT OF EXTERNAL CONTAMINANTS ON SEAL PERFORMANCE

Seal life can be adversely affected by external contaminants such as water, dust, and mud [56]. Contaminants that pass by the seal and enter the sump cavity can severely damage bearings, gears, and other machine components. Heavy duty exclusion seals can be used to prevent entry of these contaminants by facing the sealing lip toward the contaminants. These seal can be kept lubricated and clear of dirt by purging with grease; however, grease can be forced through them only if the lips point outward. These seal types (Chapter 4, Fig. 4.3e) are effective only at low speeds (less than 2.5 m/sec, 500 ft/min). If a minimal amount of contaminants is present at a seal that is retaining a fluid, a conventional secondary radial dirt lip design (Fig. 4.4b) is often used for exclusion. However, if the secondary lip contacts the shaft and runs dry at speeds over 5.00 m/sec (about 1000 ft/min), the frictional heat generated may damage both the shaft and the seal.

The results of life tests conducted using a dust box with a standard test machine are surprising (see Table 8.10). The average life of a standard seal design without a dirt lip drops from 2070 hr without dust to 1250 hr in dust. When a secondary lip is added with an interference of 0.254 mm (0.010 in.), the average life drops to 700 hr. When the secondary lip interference is increased to 0.508 mm (0.020 in.), the life is reduced even further, to only 220 hr. When helical ribs are added to the seal, life is only 240 hr without a dirt lip and 90 hr with a dirt lip.

Tests were run to understand why the addition of a contacting radial dirt lip reduced life in a dusty, high-speed environment [79]. Seals with contacting dirt lip were installed on a 127 mm (5 in.) hollow shaft, and the underlip temperature profile was measured at 1550 rpm in SAE 30 oil. The hottest

Table 8.10 Seal Life in a Dusty Environment (Arizona AC Coarse)

Material	Nitrile (NBR)
Shaft size	76.2 mm (3.000 in.)
Shaft speed	2165 rpm (8.629 m/sec, 1700 ft/min)
Sump temperature	93°C (200°F)
Lubricant	SAE 30
STBM	0.13 mm (0.005 in.)
DRO	0.13 mm (0.005 in.)
Cock	Zero
Sump level	Centerline
Cycle	20 hr on, 4 off

Seal type	Dust added?	Secondary lip configuration	Seal life
Standard	No	None	2070
	Yes	None	1250
	Yes	0.254 mm (0.010 interference)	700
	Yes	0.50 mm (0.020 interference)	220
Helix	Yes	None	240
	Yes	0.254 mm (0.010 interference)	90

Source: Chicago Rawhide.

Material	Nitrile (NBR)
Shaft size	127mm (5.000 inch)
Shaft speed	1550 RPM (10.15 m/sec, 2000 RPM
Sump temperature	93°C (200°F)
Lubricant	SAE 30
STBM	0.13mm (0.005 inch)
DRO	0.13mm (0.005 inch)
Cock	Zero
Sump level	Full
Pressure	Zero

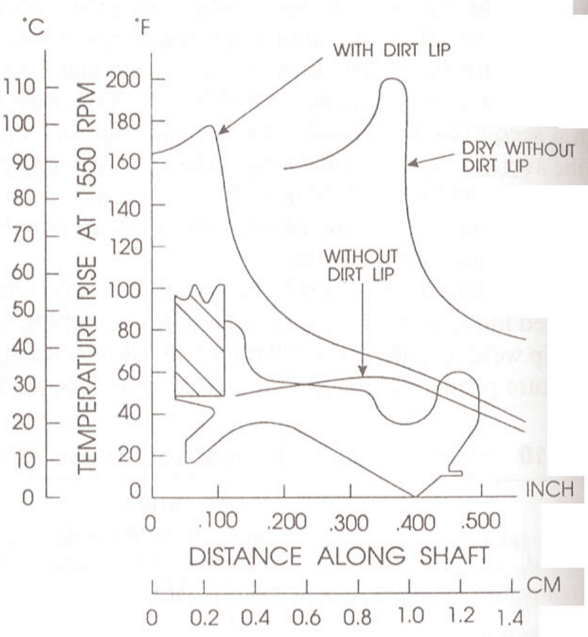


Fig. 8.17 Temperature profiles. (Courtesy of Chicago Rawhide.)

point occurred near the dry dirt lip and was 100°C (180°F), greater than the sump temperature (Fig. 8.17). When the dirt lip was removed, the primary lip ran 8.3°C (15°F) cooler than it did with the secondary lip. After 200 hr, the secondary lip had hardened and was found to be larger than shaft size.

Detailed studies were made of the failed seals from the dust tests. The failed seals without dirt lips had a wide wear track on the seal lip. The contaminants had worn out the seal, which caused the leakage, and there was a single deep groove on the shaft (Fig. 8.18a). The failed dual-lip seals did not have heavy wear on the primary lip, but the cavity between the two lips was packed with dust and the shaft surface between the two lips was heavily damaged (Fig. 8.18b, 8.18c). The reason for the leakage was clear.

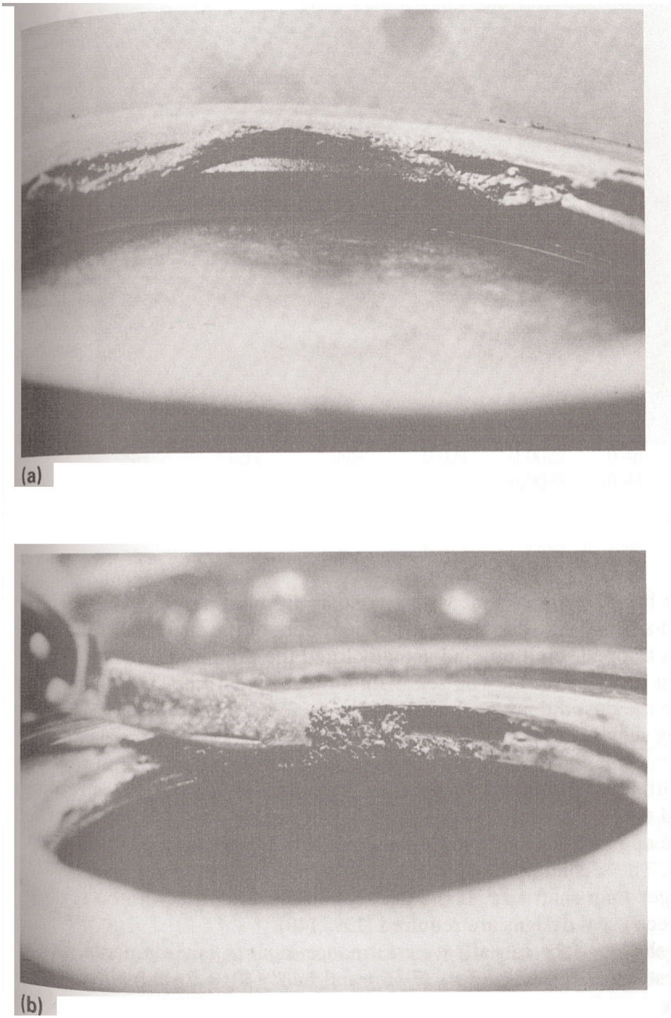


Fig. 8.18 Failed seals and shafts from dust tests. (a) Single-lip seal with worn lip. (b) Double-lip seal packed with dust. (c) Shaft heavily damaged between lips. (Courtesy of Chicago Rawhide.)

Fig. 8.18 Continued

After a brief period of time, the secondary lip became hard and slightly larger than shaft size due to the dynamic runout of the shaft. Dust entered but was trapped, built up, and actually lifted the primary lip off the shaft, allowing the oil to leak. The seal without a secondary lip had longer life because the dust did not pack into the lip area.

Many seal users specify secondary radial dirt lips for their applications. The environmental conditions for these seals can vary from clean to very dirty with shaft speeds that range from very slow to very fast. It is recommended that seals with noncontacting secondary lips be used for general-purpose applications to ensure that dust does not become trapped between the lips. The secondary lip diameter should be 0.25--0.50 mm (0.010--0.020 in.) larger than shaft size. If the dust conditions are known to be severe, then special seal designs are required [139, 140].

The shaft surface can affect performance in dirty environments. Seals were tested for 154 hr on soft (7 Rc) and hard (40 Rc) shafts in a clean environment. The shaft wear was identical. When the test was repeated in a dusty environment, the depth of the wear groove of the soft shaft was three times that of the hard shaft. Hard chrome plating over soft ground shafts had the greatest resistance to wear and also had low seal abrasive wear. Results appear in Table 8.11.

Table 8.11 Shaft and Seal Lip Wear

Material	Fluoroelastomer
Shaft size	120.65 mm (4.750 in.)
Shaft speed	2800 rpm
Sump temperature	126°C (260°F)
Lubricant	10W Delo 300
STBM	0.25 mm (0.010 in.)
DRO	0.25 mm (0.010 in.)
Cock	Zero
Sump level	Quarter shaft
Test duration	164 hr

Shaft hardness R	Dust added?	Oil Condition	Chrome on shaft	Shaft wear depth		Seal lip wear	
				mm	in.	mm	in.
7	No	Clean	No	0.0051	0.00020	0.38	0.015
40	No	Clean	No	0.0064	0.00025	0.41	0.016
7	No	Dirty	No	0.071	0.0028	0.46	0.018
40	No	Dirty	No	0.064	0.0025	0.41	0.016
7	No	Dirty	Yes	0.0064	0.00025	0.41	0.016
7	Yes	Clean	No	0.152	0.006	1.60	0.063
40	Yes	Clean	No	0.064	0.0025	0.79	0.031
7	Yes	Clean	Yes	0.0064	0.00025	0.89	0.035

Source: Chicago Rawhide.

Table 8.12 Summary of Shaft Tests

Condition	Soft shafts	Hard shafts	Chrome plating
Clean oil	Good	Good	Good
Dirty oil	High shaft wear	High shaft wear	Good
Dusty environment	High shaft and lip wear	Moderate lip and shaft wear	Moderate lip wear; shaft good

Source: Chicago Rawhide.

Tests were also carried out in a dirty oil environment. The wear depth of the soft (7 Rc) and hard (40 Rc) shafts was equivalent and quite deep (0.064 mm; 0.0025 in.). The observed shaft wear was due to etching from the chemicals contained within the dirty oil. When the soft shaft was plated with hard chrome, the depth of the shaft wear dropped to one-tenth that of the uncoated shaft (Table 8.11). The wear track width of all seals was equivalent. Conclusions are summarized in Table 8.12.

9

The Effect of Seal Design Parameters on Performance

9.1 BASELINE TEST CONDITIONS

A series of tests were conducted to determine the effect that varying seal design parameters has on shaft seal performance. The underlip temperature, torque, power consumption, and seal life were determined using the techniques described in Chapter 7. The results should be considered trends and are not intended to be representative of seals operating in field conditions. The design parameters varied during the test are shaft size, garter spring tension, primary lip sprung diameter, and R value.

The effect of a specific variable was determined by changing that variable while holding the others constant. Any deviation from the baseline conditions is noted. The elastomeric material used in the studies was nitrile (NBR). Seals used for the tests were manufactured in a single-cavity mold using NBR from a single batch of mixed material. The process parameters were held constant during the manufacturing process. Functional test parameters are listed in Chapter 8 in Table 8.1, and the values for the baseline design parameters are given in Table 9.1.

Three other special studies are described: (1) the effect of seal cross-sectional configuration and material on cold weather performance; (2) the effect of seal cross-sectional design on performance in pressure applications; and (3) the effect of secondary lip design on dust exclusion.

Table 9.1 Baseline Design Parameters

Shaft size	76.2 mm (3.000 in.)
Bore size	101.6 mm (4.000 in.)
Seal width	1.27 mm (0.500 in.)
Spring tension	2.9 N (10.5 oz)
Off-shaft R value	0.25 mm (0.010 in.)
Sprung interference	2.54 mm (0.100 in.)

Source: Chicago Rawhide.

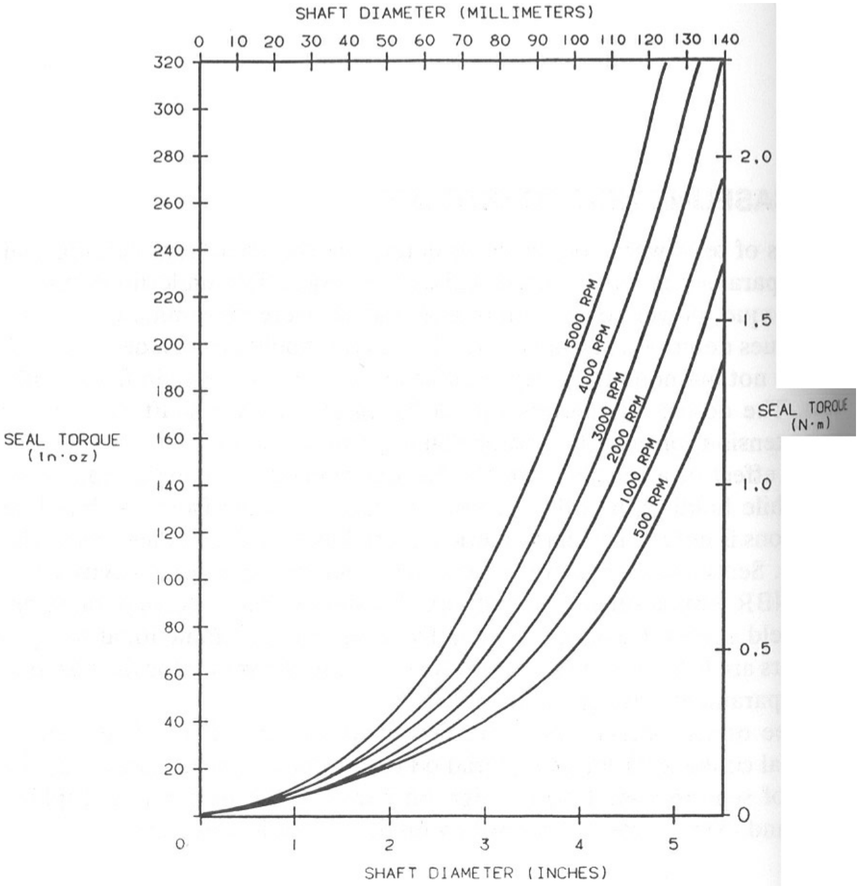


Fig. 9.1 Seal torque versus shaft diameter. (Courtesy of Rubber Manufacturers

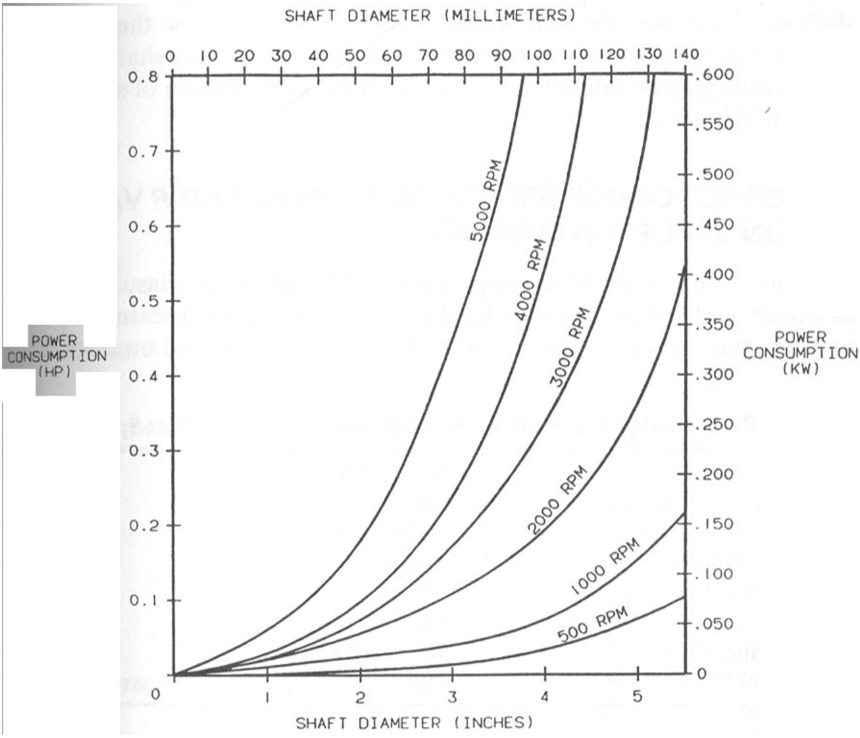


Fig. 9.2 Seal power consumption versus shaft diameter. (Courtesy of Rubber Manufacturers Association.)

9.2 EFFECT OF SHAFT SIZE ON SEAL PERFORMANCE

Changing the shaft size has a dramatic effect on radial lip seal performance [219]. At constant speed, the frictional torque and power both increase with the approximate square of the shaft diameter (Figs. 9.1 and 9.2). The frictional drag of an automotive engine rear crankshaft seal (about 90 mm shaft size) can be as much as 500 W (0.65 hp) at 5000 rpm. At 1000 rpm, the same seal will consume only 50 W (0.065 hp). Equipment designers must consider the the power losses associated with the seals and how those losses vary with speed when they are designing engines and other mechanical device. The frictional drag causes the lip temperature for larger seals to be higher than for smaller seals even when the sump temperature is the same. This increased temperature can result in a drastic decrease in seal life. Larger seals also have greater pump rates than smaller ones. Larger diameter seal are subjected to higher circumferential torque than smaller ones, so they are more subject to stick-slip. To compensate for the changes that occur as shaft size increases, the seal designer will normally increase the seal cross section. Underlip temperatures and pump rates for various shaft sizes and speeds are listed in Tables 9.2 and 9.3. Seal life as a function of shaft size is given in Table 9.4.

9.3 EFFECT OF GARTER SPRING TENSION AND R VALUE ON SEAL PERFORMANCE

Exposure to oil at elevated temperatures will cause some elastomers such as nitrile and polyacrylate to harden, take a set, and lose interference between the seal lip and the shaft. A garter spring is placed on the seal to

Table 9.2 Underlip Temperature vs. Shaft Size for Various Shaft Speeds

Material	Nitrile (NBR)
Lubricant	SAE 30
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Sump temperature	93°C (200°F)
Sump level	Full

			Underlip temperature (°C)			
Shaft size	500 rpm	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
50.8 mm (2.000 in.)	98	101	108	114	120	124
76.2 mm (3.000 in.)	100	103	112	118	123	128
95.25 mm (3.750 in.)	102	109	117	123	128	--
130.18mm (5.125 in.)	104	111	120	126	133	--

Source: Chicago Rawhide.

Table 9.3 Seal Pump Rate vs. Shaft Size for Various Shaft Speeds[a]

			Pump rate (cm^3/min)			
Shaft size	500 rpm	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
31.8 mm (1.250 in.)	--	0.005	0.015	0.030	0.040	0.055
50.8 mm (2.000 in.)	0.020	0.025	0.040	0.052	0.070	0.080
66.7 mm (2.625 in.)	0.025	0.040	0.075	0.120	0.155	0.190
76.2 mm (3.000 in.)	0.030	0.045	0.150	0.138	0.180	0.225
95.25 mm (3.750 in.)	0.035	0.065	0.120	0.180	0.24	0.30
130.18 mm (5.125 in.)	0.150	0.200	0.310	--	--	--

[a] Conditions same as for Table 9.2.

Source: Chicago Rawhide.

Table 9.4 Seal Life vs. Shaft Size

Material	Nitrile (NBR)
Lubricant	SAE 30
Shaft speed	8.63 m/sec (1700 ft/min)
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Sump temperature	93°C (200°F)
Sump level	Shaft centerline
Cycle	20 hr on, 4 hr off

Template no.	Shaft size	Average seal life (hr)
1	31.75 mm (1.250 in.)	1381
2	50.8 mm (2.000 in.)	1945
3	76.2 mm (3.000 in.)	2071
4	94.25 mm (3.750 in.)	2664
5	130.18 mm (5.125 in.)	2500

Source: Chicago Rawhide.

Table 9.5 Underlip Temperature vs. Spring Tension for Various Shaft Speedsa

Material	Nitrile (NBR)
Lubricant	SAE 30
Shaft size	76.2 mm (3.000 in.)
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Sump temperature	93°C (200°F)
Sump level	Full

Spring tension	Underlip temperature (°C)					
	500 rpm	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
Zero (no spring)	96	98	100	106	114	116
2.9 N (10.5 oz)	98	99	113	123	127	128
4.2 N (15.0 oz)	100	103	117	125	131	134
5.6 N (20.0 oz)	101	108	123	133	137	143

Source: Chicago Rawhide.

Table 9.6 Seal Torque vs. Spring Tension for Various Shaft Speeds[a]

Spring tension	Seal torque (N-m)					
	500 rpm	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
Zero (no spring)	0.15	0.18	0.24	0.31	0.38	0.45
2.9 N (10.5 oz)	0.28	0.30	0.38	0.45	0.53	0.60
4.2 N (15.0 oz)	0.34	0.38	0.45	0.52	0.60	0.68
5.6 N (20.0 oz)	6.42	0.46	0.55	0.65	0.73	0.82

[a] Conditions as for Table 9.5.

Source: Chicago Rawhide.

Table 9.7 Seal Power Consumption vs. Sprung Interference for Various Shaft Speeds[a]

	Power consumption (W)					
Spring tension	500 rpm	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
Zero (no spring)	15	20	55	100	175	240
2.9 N (10.5 oz)	30	40	95	150	220	310
4.2 N (15.0 oz)	35	50	110	180	260	340
5.6 N (20.0 oz)	40	75	130	215	320	450

[a] Conditions as for Table 9.5.

Source: Chicago Rawhide.

compensate for these material changes, improve seal reliability, and extend the life of the seal. Increasing the spring tension will increase the lip tension, frictional torque, and power consumption (Tables 9.5--9.7). The average seal life without a spring is less than 500 hr. The addition of a spring doubles seal life, and the life is relatively insensitive to the magnitude of the spring tension. The seal lip wear width does not change appreciably with spring tension (Table 9.8). It is good design practice to select a spring tension that will provide good sealing without generating excessive frictional torque as the latter leads to excessive power losses. High spring tensions may also reduce seal life at higher sump temperatures. The oil viscosity will drop as lip temperature rises, which will result in a thinner lubricating film, heavy wear, oil coking, and reduction in life.

Material selection will also affect the amount of spring tension required. When exposed to oil, silicone materials swell much more than nitriles, polyacrylates, and fluoroelastomers. Greater spring tensions are required tor high-swell materials to maintain the desired interference levels and radial load. Materials such as PTFE that do not harden or swell in hot oil may not require a spring at all. The seal designer and user must work together to obtain the longest seal life with the best reliability at the lowest level of power consumption.

Variation in spring position or R value does not have a significant effect on uniderilip temperature, seal torque, power consumption, or pump rate as long as the R value is positive. The R value is defined as the distance between the lip contact point and the spring centerline. If the spring centerline is on

Table 9.8 Seal Life and Lip Wear Width vs. Spring Tension

Material	Nitrile (NBR)
Lubricant	SAE 30
Shaft size	76.2 mm (3.000 in.)
Shaft speed	2165 rpm, 8.63 m/sec (1700 ft/min)
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Sump temperature	107°C (225°F)
Sump level	Shaft centerline
Cycle	20 hr on, 4 hr off

Spring tension	Seal life (hr)	Wear width
Zero (no spring)	479	0.20--0.30 mm (0.008--0.012 in.)
2.9 N (10.5 oz)	924	0.71--0.97 mm (0.028--0.038 in.)
42 N (15.0 oz)	969	0.50--0.94 mm (0.020--0.037 in.)
5.6 N (20.0 oz)	1124	0.74--1.00 mm (0.029--0.040 in.)

Source: Chicago Rawhide.

the air side of the contact point, the R value is defined to be positive. The R value is negative if the spring centerline is on the oil side of the contact point. Spring position alters the pressure distribution in the seal lip--shaft contact area. Changing the R value from zero to +0.76 mm (+0.030 in.) had no effect on seal life or lip wear width. When the R value became negative, instant leakage was observed.

9.4 EFFECT OF SPRUNG INTERFERENCE ON SEAL PERFORMANCE

The ultimate life of nitrile seals occurs when the elastomer becomes hard, loses lip-to-shaft interference, takes a set, and cannot follow shaft eccentricity. Increasing the initial or design sprung interference will increase seal life, but lip and shaft wear will also increase. The increase in seal life with sprung interference tends to level out, but lip wear continues to increase (Table 9.9). Seal load also increases and the lubricating film thickness decreases as the sprung interference increases. As a result, lip temperature, torque, and power consumption all increase with increasing sprung interference (Tables 9.10--9.12).

Table 9.9 Seal Life and Lip Wear vs. Sprung Interference

Sprung interference	Seal life (hr)	Wear width
1.52 mm (0.060 in.)	310	0.43--0.56 mm (0.017--0.022 in.)
2.03 mm (0.080 in.)	400	0.48--0.66 mm (0.019--0.026 in.)
2.54 mm (0.100 in.)	480	0.50--0.76 mm (0.020--.030 in.)
3.05 mm (0.120 in.)	510	0.71--1.02 mm (0.028--0.040 in.)
3.56 mm (0.140 in.)	530	0.76--1.27 mm (0.030--0.050 in.)
4.06 mm (0.160 in.)	540	1.40--2.41 mm (0.055--0.045 in.)

[a] Conditions as for Table 9.8, except that sump temperature was 121°C (250°F).

Source: Chicago Rawhide.

Table 9.10 Underlip Temperature vs. Sprung Interference for Various Shaft Speeds

Material	Nitrile (NBR)
Shaft size	76.2 mm (3.000 in.)
Lubricant	SAE 30
DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Seal cock	Zero
Pressure	Zero
Sump temperature	93°C (200°F)
Sump level	Full

	Underlip temperature (°C)				
Sprung interference	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
1.52 mm (0.060 in.)	6.0	11.5	15.5	19.0	20.5
2.03 mm (0.080 in.)	6.0	12.0	17.0	20.5	22.0
2.54 mm (0.100 in.)	6.5	12.5	17.5	22.0	24.5
3.05mm (0.120 in.)	6.5	13.0	18.5	23.0	26.0
3.56 mm (0.140 in.)	7.0	14.5	20.0	24.0	27.5
4.06 mm (0.160 in.)	7.5	15.0	21.0	25.5	28.5

Source: Chicago Rawhide.

Table 9.11 Seal Torque vs. Sprung Interference for Various Shaft Speeds[a]

	Seal torque (N-m)				
Sprung interference	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
1.52 mm (0.060 in.)	0.195	0.266	0.320	0.400	0.455
2.03 mm (0.080 in.)	0.200	0.275	0.330	0.410	0.480
2.54 mm (0.100 in.)	0.205	0.280	0.350	0.420	0.490
3.05 mm (0.120 in.)	0.215	0.300	0.375	0.440	0.515
3.56 mm (0.140 in.)	0.245	0.320	0.400	0.470	0.540
4.06 mm (0.160 in.)	0.250	0.330	0.415	0.490	0.555

[a] Conditions as for Table 9.10.

Source: Chicago Rawhide.

Table 9.12 Seal Power Consumption vs. Sprung Interference for Various Shaft Speeds[a]

	Power consumption (W)				
Sprung interference	1000 rpm	2000 rpm	3000 rpm	4000 rpm	5000 rpm
1.52 mm (0.060 in.)	20.4	55.6	100.4	167.4	238.0
2.03 mm (0.080 in.)	20.9	57.5	103.6	171.5	251.0
2.54 mm (0.100 in.)	21.4	58.6	109.8	175.7	256.3
3.05 mm (0.120 in.)	22.5	62.8	117.7	184.1	269.3
3.56 mm (0.140 in.)	25.6	66.9	125.5	196.6	282.4
4.06 mm (0.160 in.)	26.2	69.0	130.2	205.0	290.3

[a] Conditions as for Table 9.10.

Source: Chicago Rawhide.

The optimum diametrical interference level for standard nitrile seals is about 2.5 mm (0.100 in.) for a 76.2 mm (3.000 in.) shaft size and varies with shaft size. Seal users and designers may select premium materials that are not appreciable affected by exposure to hot oils so the sprung interference can be reduced. Reduction in spring interference will reduce seal torque, power consumption, and heat generation.

9.5 SEAL DESIGN AND MATERIAL CONSIDERATIONS FOR LOW-TEMPERATURE OPERATION

The exposure of machinery to subarctic temperatures has become more frequent within recent years, and special lubricants have been developed to cope with these hostile conditions. Cold temperatures harden most elastomeric materials and make them brittle enough to crack when subjected to impulse forces. The temperature at which cracking occurs is dependent on the dynamic runout of the shaft. As the runout increases, the force required to move the sealing lip at start-up increases dramatically, and fracture will occur at higher temperatures.

Three seal templates were studied to determine how seal design affects the point at which fracture occurs. Seals for a 76.2 mm (3.000 in.) shaft were made from three different polyacrylate materials with a standard template, a template with a long, thin section to improve flexibility and a short, thick section that reduces flexibility. The seals were installed in a test head without lubricants and with a known dynamic runout. The head was placed in an environmental chamber and cooled to the desired temperature. After the system was soaked for 12 hr, the shaft was rotated at 60 rpm. The seals were inspected for cracks. If no cracks occurred, the temperature was lowered and the test was repeated until fracture occurred. It is recommended that some mechanism to drive the shaft uniformly without removing the head from the environmental chamber be developed to ensure test consistency. The results of the test appear in Table 9.13. The fracture point of the material was independent of the seal design but highly dependent on material formulation. Compound C was the poorest and A was the best. At a runout of 0.76 mm (0.030 in.), the fracture of C occurred at --18°C (0°F). Compound A did not fracture until the test equipment was cooled to --40°C (--40°F).

Seal leakage can occur at low temperatures even if the lip does not fracture. Material modulus increases as temperature decreases. The lip becomes stiff and may be unable to follow shaft eccentricities. This may not be a problem with conventional fluids because they become very viscous at low temperatures and will not leak through the small gap between the seal lip and the shaft. Synthetic fluids do not thicken at low temperatures, and leakage may occur. Friction will warm the elastomeric lip, followability of shaft eccentricity will return, and leakage will stop. The high material modulus at low temperatures can cause excessive wear because the radial load between the seal and the shaft increases. If the seal case and housing bore are made of different materials, then differential thermal contraction can result in bore leakage. If the seal is subjected to temperatures of --20°C (-4°F) or lower, then the low-temperature properties of the elastomer must be considered. The seal case material should be the same as the bore material to minimize the effect of thermal contraction at low temperatures If this is not possible, rubber O.D. seals should be used.

9.6 THE EFFECT OF SEAL CROSS-SECTIONAL DESIGN IN PRESSURE APPLICATIONS

Ten seal designs were tested under constant pressure of 690 kPa (100 psi) at a shaft speed of 5.076 m/sec (1000 ft/min). The shaft was hardened to Rc 60 and had a dynamic runout of 0.13 mm (0.005 in.) and an STBM of 0.13 mm (0.005 in.). The sump temperature was 93°C (200°F), and the lubricant was

Table 9.13 Cold Fracture of Polyacrylate Seals

	Temperature at which fracture occurs (°C)				
	Compound A			Compound B	Compound C
Dynamic runout	Thick flex section	Thin flex section	Standard flex section	Standard flex section	Standard flex section
0.254 mm (0.010 in.)	--44.4	--45.5	--45.5	--40	--31.7
0.381 mm (0.015 in.)	--42.8	--42.8	--42.8	--37.2	--26.1
0.508 mm (0.020 in.)	--37.2	--40.0	--40.0	--34.4	--23.3
0.635 mm (0.025 in.)	--34.4	--40.0	--37.2	--31.7	--20.6
0.762 mm (0.030 in.)	--37.2	--37.2	--37.2	--31.7	--17.8
0.889 mm (0.035 in.)	--37.2	--36.7	--37.2	--30.0	--17.8
1.016 mm (0.040 in.)	--34.4	--37.2	--37.2	--28.9	--17.8

Source: Chicago Rawhide.

SAE 10W engine oil. The test cycle was 20 hr at speed with heat and pressure applied and a 4 hr down cycle with no heat, pressure, or shaft rotation. Only one seal design had an average life longer than 40 hr (Table 9.14). This seal (design A) had a life of 226 hr and used a thermoplastic backup ring and seal support that was snapped onto the seal's back face. Several of the other designs are typical pressure designs for pumps rated at 690 kPa (100 psi), but they failed earlier than expected. The pressure in the seal area in the actual pump is actually much less than 690 kPa (100 psi). Pressure probes should be used in tests of actual pump applications to determine what the pressure actually is in the seal region when selecting a design. The pressure rating for a seal design should not be the same as the pump rating unless these measurements show that the seal region has the same pressure rating as the pump.

9.7 THE EFFECT OF SECONDARY LIP DESIGN ON SEAL PERFORMANCE IN DUST ENVIRONMENT

External contaminants can reduce seal life and enter machine elements, which may result in severe damage to bearings and other internal mechanical components. Many types of secondary sealing lips have been offered by seal manufacturers to prevent ingress of contaminants. The effectiveness of some of these designs has been questioned over the years [79]. Some designs have contacting lips that have little or no lubrication. Heat is generated that may reduce life. A vacuum may be formed between the contacting secondary lip and the primary lip, which may increase lip load and seal wear.

Peisker and Sahiouni [139, 140] conducted a series of tests in dust to evaluate a common primary lip design made of fluoroelastomer combined with various types of excluder lips to determine which design was the most effective. A test apparatus (Fig. 9.3) was used to determine how much (if any) vacuum was developed between the two contacting lips. The vacuum port in the spindle is adjusted to be between the primary and secondary lips. The test was run at 1700 rpm with a sump temperature of 107°C (225°F) (Table 9.15). The lubricant was SAE 30 at centerline. The vacuum was recorded on a strip recorder for a period of 20 hr. The vacuum reached a maximum shortly after start-up for most designs but decayed to a low level within 20 hr. The vacuum disappeared during the 4 hr shutdown period, and only nominal levels (0.3--0.7 kPa; 0.1--0.2 in. Hg) were generated during the second running period. A grease pack between the lips made no difference in the amount of vacuum formed. It was concluded that vacuum buildup between the lips has a negligible effect on seal load and performance.

Table 9.14 Life of Seal Design Variations Under 690 kPa (100 psi) Pressure--Shaft Hardened to Rc 60

DRO	0.13 mm (0.005 in.)
STBM	0.13 mm (0.005 in.)
Shaft speed	5.076 m/sec (1000 ft/min)
Sump temperature	93°C (200°F)
Lubricant	SAE 10W
Pressure	690kPa (100 psi)
Cycle	20 hr on, 4 hr off

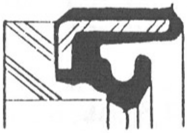
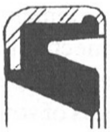
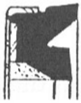
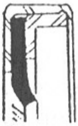
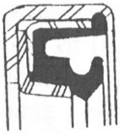
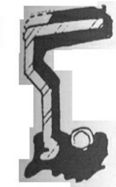
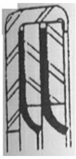
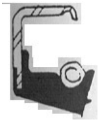
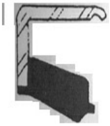
Seal cross section	Shaft diameter	Seal life (hr)	Seal material
A 	25 mm (0.984 in.)	226	Nitrile (plastic backup)
B 	26.97 mm (1.062 in.)	40	Nitrile
C 	30.15 mm (1.187 in.)	27	Nitrile (plastic backup)
D 	25.4 mm (1.000 in.)	23	PTFE
E 	31.75 mm (1.250 in.)	21	Nitrile (steel backup)

Table 9.14 Continued

Seal cross section	Shaft diameter	Seal life (hr)	Seal material
F 	20.62 mm (0.812 in.)	20	Nitrile (curled steel backup)
G 	19.05 mm (0.750 in.)	19	PTFE
H 	22.2 mm (0.875 in.)	19	Nitrile
I 	31.75 mm (1.250 in.)	17	Nitrile

Source: Chicago Rawhide.

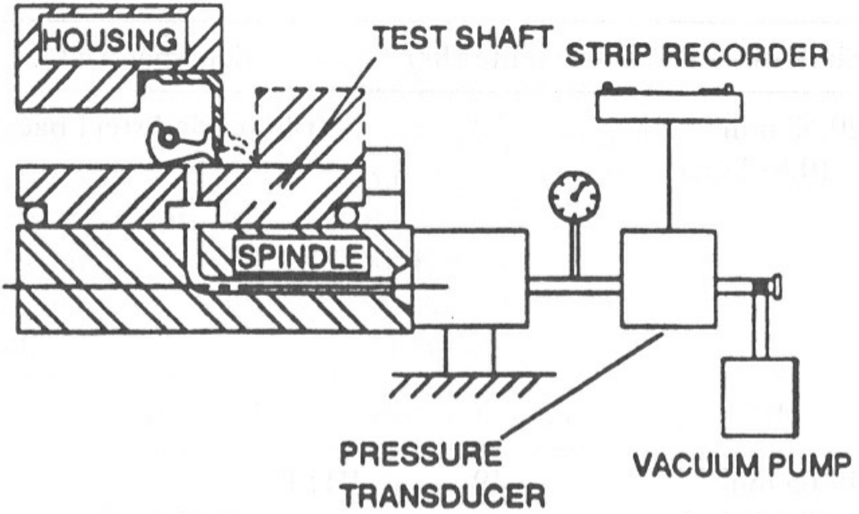

HOUSING
TEST SHAFT
STRIP RECORDER
SPINDLE
PRESSURE TRANSDUCER
VACUUM PUMP

Fig. 9.3 Vacuum test apparatus. (Courtesy of Chicago Rawhide.)

Table 9.15 Test Parameters for Dust and Vacuum Tests

Dust box	Cylindrical cavity, 203 mm (8 in.) diam, 203 mm (8 in.) wide
Dust paddle	104 mm (5.525 in.) diameter
Paddle speed	700 rpm
Dust	AC coarse
Amount of dust	800 mi
Seal	Various
Shaft diameter	76 mm (3.000 in.)
Shaft speed	2165 rpm (1700 ft/min; 10 m/sec)
DRO	0.13 mm (0.005 in.) TIR
STBM	0.25 mm (0.010 in.) TIR
Fluid temperature	107°C (225°F)
Test fluid type	SAE 30
Fluid level	Shaft centerline

Source: Chicago Rawhide.

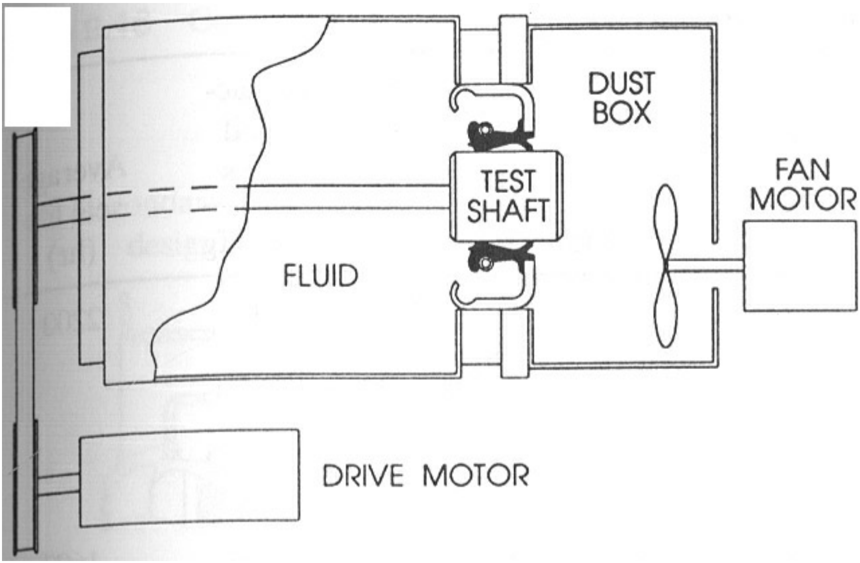


Fig. 9.4 Dust box for functional tests. (Courtesy of Chicago Rawhide.)

The dust tests were run with a dust box added to a conventional test machine (Fig. 9.4). The results of the vacuum and dust life tests appear in Table 9.16. Analysis of the data reveals some interesting results. The axial dirt lip has the longest life in dust. The radial flange that the axial dirt lip contacts also acts as a flinger to push dust away from the seal as the flinger rotates. The configuration that has the second longest life is one without any dust lip. All other radial lip configurations have lives shorter than that of the seal without a dirt lip. Failure occurs when the dust collects in the cavity between the lips and impedes seal function (Fig. 8.18). The noncontacting radial lip has a longer life than the contacting secondary radial lip. The gap allows some dust to escape, and the dust buildup between the lips takes longer than with the contacting lip. The nylon felt lip gave the worst performance. A Weibull chart comparing the performance of the axial lip, the contacting radial lip, and no dust lip shows the axial lip to have the greatest reliability as the Weibull slope is 4.57 (Fig. 9.5). The slope of the contacting radial lip is only 1.90, which is about the same as for seals with no exclusion lips (1.97).

The conclusion is: Do not use any radial dirt lip design in dust at elevated shaft speeds. The axial dirt lip gives the longest life and is recommended for high-speed, dusty applications. Care must be taken when installing the axial unitized seal. The recommended installation tool is shown in Fig. 9.6.

Table 9.16 Evaluation of Secondary Lips

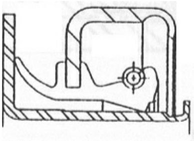
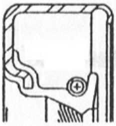
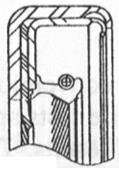
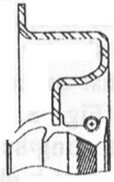
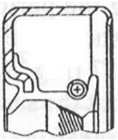
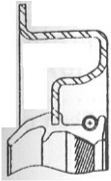
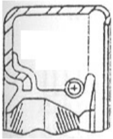
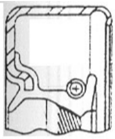
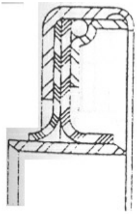
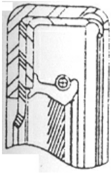
Secondary lip design	Description	Maximum vacuum formed between lips		Average seal life (hr)
		kPa	in. Hg	
	Axial	0	0	2200
	No secondary dirt lip	--	--	1607
	Inverted PTFE with spiral	2.7	0.8	1266
	Nonvented dual extended	1.0	0.3	1217
	Noncontacting radial --	--	678	

Table 9.16 Continued

Secondary lip design	Description	Maximum vacuum formed between lips		Average seal life (hr)
		kPa	in. Hg	
	Vented dual - extended	-	--	672
	Contacting radial with helix	0	0	477
	Contacting radial	0	0	422
	PTFE radial	2.4	0.7	349
	Nylon felt radial	0	0	298

Source: Chicago Rawhide.

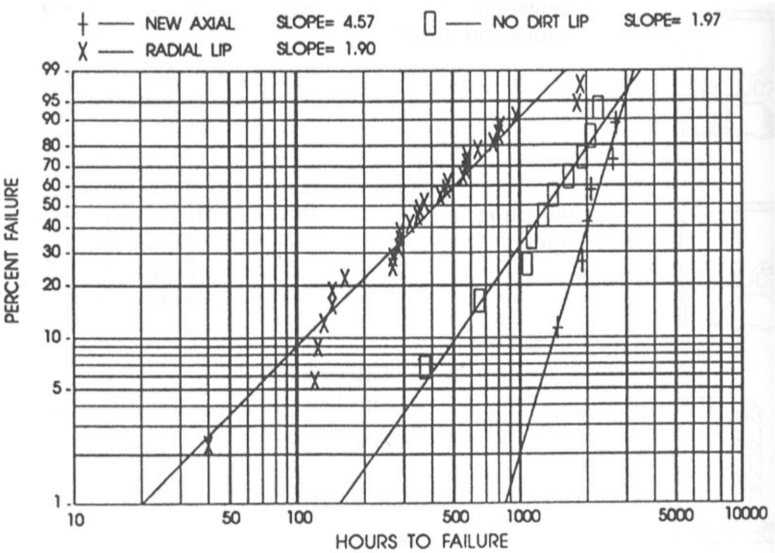


Fig. 9.5 Seal life in a dust environment. (Courtesy of Chicago Rawhide.)

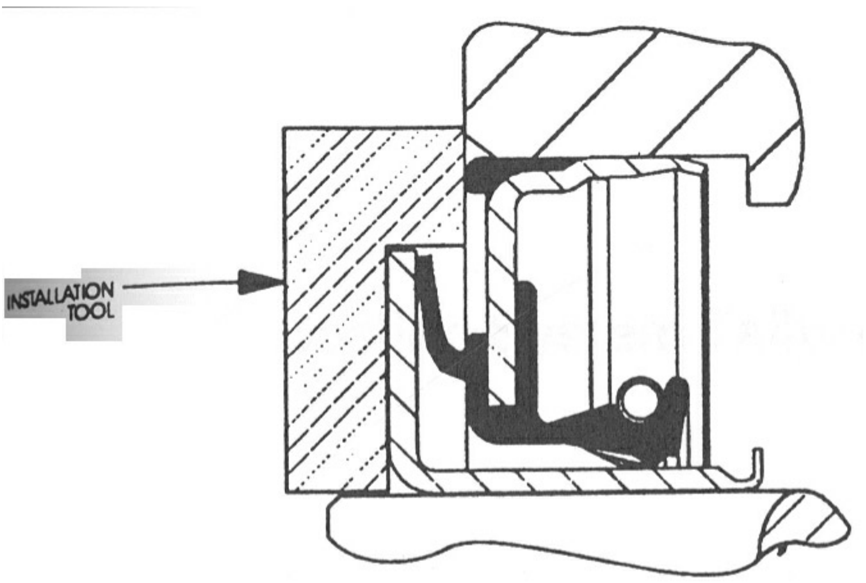

INSTALLATION TOOL

Fig. 9.6 Recommended installation tool and procedure for unitized axial dirt lip. (Courtesy of Chicago Rawhide.)

10

Sealing System Failure Analysis

10.1 THE SEALING SYSTEM

When a sealing system fails, very often the seal is removed and returned to the manufacturer with a request that the company explain why the leakage occurred. This is often an impossible task because evidence that could define the root cause of the failure is not included or is destroyed when the seal is removed. The seal is only one part of the sealing system. The shaft or running surface, the housing bore, the seal, and the lubricant condition must all be carefully and systematically examined to determine the real cause of failure to determine what corrective action should be taken. A check list should be employed to lead an investigator through a sequential check of the sealing system and its immediate environment with the seal in place, an examination of the seal after removal, and a study of the shaft, housing, and lubricant after the seal is removed [58, 59]. The Seal Technical Committee of the Rubber Manufacturers Association [166] has prepared check lists that should be used to help determine the cause of leakage (Tables 10.1--10.3). A shorter version of the three check lists has been prepared for use in the field (Table 10.4).

Table 10.1 Check List for the Examination of the Sealing System and Immediate Environment with the Seal in Place

1. Record seal application, equipment identification, miles or hours of operation, and type of complaint.
2. Record leakage amount.
   - ☐ Slight leakage
   - ☐ Immediate area damp
   - ☐ Heavy leakage
3. Record source of leakage.
   - ☐ Between shaft and seal lip
   - ☐ Between O.D. of seal and bore
   - ☐ At retainer bolt holes
   - ☐ At retainer gasket
   - ☐ Between wear sleeve and shaft
   - ☐ Through seal on assembled seals
4. Record condition of surrounding area.
   - ☐ Seal area clean
   - ☐ Mud or dust packed in seal area
5. Clean surrounding area and inspect.
   - ☐ Nicks on bore chamfer
   - ☐ Seal loose in bore
   - ☐ Paint spray on seal lip
   - ☐ Seal cocked in bore (amount) _________
   - ☐ Seal installed in incorrect orientation (backward)
   - ☐ Seal case deformed
   - ☐ Shaft-to-bore misalignment
6. Rotate shaft if possible and check for runout and end play.
   - ☐ Excessive shaft end play (amount)_________
   - ☐ Excessive shaft runout (amount)__________
7. Confirm leakage location.

   If location of leakage cannot be confirmed at this point, either introduce ultraviolet dye into the sump or spray area with white powder, operate for 15 min, and check for leakage with ultraviolet or regular light.
8. Remove seal.

   When above analysis is complete, mark the seal at the 12 o'clock position and remove it carefully from the application. Record oil type and obtain used sample from application.

Source: Rubber Manufacturers Association.

Table 10.2 Check List for Seal Examination After Removal

1. Clean the removed seal in a mild solvent. Do not attempt to scrape away carbon, etc. Inspect the seal using this checklist.
2. Check primary lip area.
   - ☐ Normal wear
   - ☐ No wear
   - ☐ Excessive wear
   - ☐ Eccentric wear
   - ☐ Inverted lip due to poor installation
   - ☐ Nicks, scratches, or cuts at lip contact area
   - ☐ Hardened or cracked rubber
   - ☐ Coked oil on lip
   - ☐ Softening or swelling
3. Check seal outside diameter.
   - ☐ Normal
   - ☐ Severe axial scratches
   - ☐ Peeled rubber
   - ☐ Hardened rubber
   - ☐ Nonfills or cuts
4. Check spring and spring groove area.
   - ☐ Spring normal and in place
   - ☐ Spring missing
   - ☐ Spring corroded
   - ☐ More than one spring
   - ☐ Separated spring
5. Record the following measurements.

   Primary lip inside diameter (_______)

   Primary lip radial force (_______)

   Seal outside diameter (_______)

   Spring inside diameter (_______)

   Spring tension (_______)

   Primary lip wear band width

   Min. (_______)

   Max. (_______)

Source: Rubber Manufacturers Association.

Table 10.3 Check List for Examination of Shaft Housing and Lubricant After Seal Removal

1. Inspect housing bore area.
    - ☐ Measure bore diameter (_______)
    - ☐ Bore chamfer damaged
    - ☐ Flaws or voids in housing
    - ☐ Tool withdrawal marks in bore
    - ☐ Bore surface scratched or galled
2. Inspect the shaft wear track.
    - ☐ Measure shaft diameter (_______)
    - ☐ Shaft surface corroded
    - ☐ Seal wear path in wrong location
    - ☐ Scratches or nicks at lip contact area
    - ☐ Measure wear path width (_______)
    - ☐ Discoloration on shaft surface
    - ☐ Coked lubricant present
    - ☐ Shaft chamfer damaged or missing
    - ☐ Wear sleeve loose on shaft (if applicable)
3. Measure shaft characteristics (if possible).
    - ☐ Measure surface roughness (______Ra)
    - ☐ Measure depth of wear path (_______)
    - ☐ Measure shaft lead (_______deg)
    - ☐ Measure shaft hardness (_______Rc)
    - ☐ Check for proper shaft material
4. Inspect and compare properties of used application lubricant to new lubricant.
    - ☐ Contaminants in filtered lube
    - ☐ Color different from new lube
    - ☐ Viscosity different from new lube
    - ☐ Odor different from new lube

Source: Rubber Manufacturers Association.

Table 10.4 Sealing System Leakage Analysis Short Form[a]

Seal Application:		Equipment Identification:	
Miles/Hours of Operation:	Complaint:		
STEP 1. Inspect the seal application before removal.			
Amount of Leakage	1 Slight	1 Seal Area Damp	1 Heavy Leakage
Condition of Area	☐ Clean	☐ Dusty	☐ Mud Packed
Leakage Source	☐ Between Lip & Shaft	☐ Between OD & Bore	
	☐ At Retainer Gasket	☐ Between Elements of Seal	
	☐ At Retainer Bolt Holes	☐ Between Wear Sleeve & Shaft	
STEP 2. Wipe area clean and inspect.			
Check	☐ Nicks on Bore Chamfer	☐ Seal Loose in Bore	
Conditions	☐ Seal Cocked in Bore	☐ Seal Case Deformed	
Found	☐ Seal Installed Improperly	☐ Paint Spray on Seal	
	☐ Shaft to Bore Misalignment	☐ Other ____________________	
STEP 3. Rotate shaft if possible.			
Check			
Conditions	☐ Excessive End Play	☐ Excessive Runout	
STEP 4.	If the location of leakage cannot be confirmed at this point, either introduce ultraviolet dye into the sump or spray area with white powder, operate for 15 minutes and check for leakage with ultraviolet or regular light.		
STEP 5.	Mark the seal at the 12 o'clock position and remove it carefully.		
	☐ Retain an Oil Sample		
	☐ Retain an Oil Sample		
STEP 6. Inspect the application with seal removed.			
Check	☐ Rough Bore Surface	☐ Flaws or Voids in Bore	
Conditions	☐ Shaft Clean	☐ Shaft Corroded	
Found	☐ Coked Lube on Shaft	☐ Shaft Discolored	
	☐ Shaft damaged		
STEP 7. Inspect the seal.			
Primary	☐ Normal	☐ Excessive	☐ Eccentric
Lip Wear	☐ None		
Primary	☐ Normal	☐ Damaged	☐ Hardened (Stiff)
Condition	☐ Soft (Flexible)		
Seal O.D.	☐ Normal	☐ Axial Scratches	☐ Damaged Rubber
Spring	☐ In Place	☐ Missing	☐ Separated
	☐ Corroded		
Comments:			
Completed By: __________________		Date: ___________________	

[a] Intended for field or shop use where the more comprehensive three-part checklist may not be practical.

Source: Rubber Manufacturers Association.

10.2 LEAKAGE ANALYSIS: BEFORE SEAL REMOVAL

10.2.1 Bore Inspection Before Seal Removal

Before the seal is removed from the bore, the bore should be checked for any nicks or burrs on the bore chamfer that may have damaged the seal O.D. and caused leakage. The seal should be checked to ensure that it is not loose in the bore. See Table 10.5 for possible defects, probable causes, and corrective action.

Table 10.5 Bore Inspection Before Equipment Disassembly

Observation or defect	Probable cause	Corrective action
Nick on bore chamfer	Mishandling prior to seal installation	Review bore/housing machining and handling practices.
Tool chatter on chamfer surface	Improper feeds, speeds, coolant, or tool configuration	Review and correct machining . practices
Insufficient material removal; chamfer not adequate	Wrong dimensions on print; poor machining practices	Check casting dimensions for proper material allowance; check machining locations for proper gauge points.
Seal loose in bore	1. Oversize bore I.D.	1. Check bore machining dimensions for out-of tolerance condition.
	2. Undersize seal O.D.	2. Check seal O.D. for out-of-tolerance condition.
	3. Rolling of seal into bore during installation	3. Review installation procedure and use proper installation tools.
	4. Bore sizing	4. Increase bore material hard ness or use bore sealant.
	5. Excessive shrinkage/ hardening of rubber O.D. seal	5. Review application tempera tures and seal material specifications.
	6. Deformation of seal during installation	6. Review installation procedure and use of proper tool.

Source: Rubber Manufacturers Association.

10.2.2 Seal Inspection Before Seal Removal

The seal should be inspected closely before being removed from the application. Improper installation is the major cause of early leakage. If the seal is removed before a thorough inspection has been conducted, then clues that would identify improper installation as a problem will be destroyed. Bore leaks can often be identified before the seal is removed, leading to a careful inspection of both the housing and the seal O.D. after the seal is removed. Occasionally, a seal will be installed backward, which will result in instant catastrophic leakage. A backward installation is difficult to diagnose once the seal has been removed. Table 10.6 and Fig. 10.1 list and

Table 10.6 Seal Inspection Before Equipment Disassembly

Observation or defect	Probable cause	Corrective action
Paint on seal lip	1. Lack of paint mask	1. Review painting procedure and recommend a mask.
	2. Service or in-field paint procedure	2. Issue a service bulletin to prevent paint over-spray or specify a mask.
Seal cocking	1. Seal installation (Fig. 10.1a)	1. Use proper installation tool; check installation force to ensure complete installation.
	2. Insufficient or improper bore chamfer	2. Provide proper amount and lead-in angle for chamfer.
	3. Excessive seal interference with rubber O.D. seal	3. Check bore I.D. and seal O.D. for proper dimensions.
Improper installation	1. Backward installation caused by lack of proper installation tool or visual aid (Fig. 10.1b)	1. Provide foolproof installation tool and/or visual aid to identify proper orientation.
	2. Improper axial location of seal (Fig. 10.1c)	2. Provide proper installation too
	3. Improper axial position shaft (Fig. 10.1d)	3. Provide proper installation of tool and visual aid for proper position.

Table 10.6 Continued

Observation or defect	Probable cause	Corrective action
Leakage at seal O.D.	1. Oversize bore/ undersize seal	1. Check bore and seal diameters at removal.
	2. Damaged housing	2. Check upon removal.
	3. Damaged seal	3. Check for O.D. damage upon removal.
	4. Differential thermal expansion (aluminum or magnesium housing)	4. Calculate fit at maximum temperature.
Seal case deformation	1. Dented heel face caused by hammer installation. (Fig. 10.1e)	1. Provide proper installation tool.
	2. Dished heel face caused by improper installation tool (Fig. 10.1f)	2. Provide proper installation tool.
	3. Severe dish or bulge of seal case outer face that occurs at time of assembly due to excessive interference between seal O.D. and bore	3. Reduce interference between seal O.D. and housing bore.
Leakage between components of assembled seal	1. Insufficient clamping force by seal manufacturer	1. Check to see if seal element is loose after disassembly. Consult with seal manufacturer.
Excessive contaminants packed in seal area	1. Lack of protective shield or slinger	1. Redesign area to shield seal.
	2. Failure of auxiliary lip	2. Look for dust packing between secondary and primary lips after disassembly. Look for cuts, damage, or high wear on auxiliarly lip. Change auxiliary lip design.

Source: Rubber Manufacturers Association.

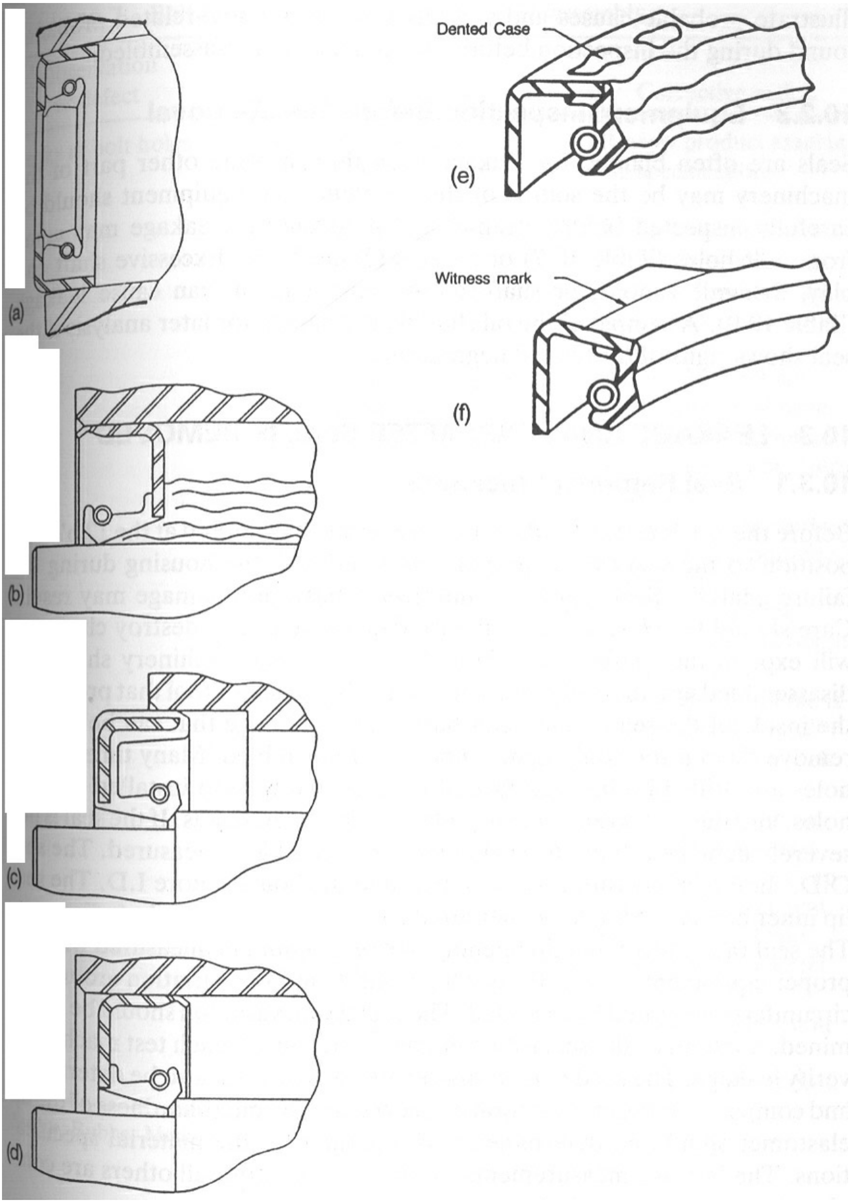

Dented Case
(e)
Witness mark
(f)
(a)
(b)
(c)
(d)

Fig. 10.1 Seal inspection before equipment disassembly. (a) Cocked seal. (b) Seal installed backward. (c) Improper axial location of seal. (d) Improper axial position of shaft (e) Dented case (hammer blows). (f) Dished case. (Courtesy of Rubber Manufacturers Association.)

illustrate probable causes and corrective action for seal-related problems found during the inspection before the equipment is disassembled.

10.2.3 Equipment Inspection Before Seal Removal

Seals are often blamed for leakage even though some other part of the machinery may be the source of the problem. The equipment should be carefully inspected before cleanup and disassembly. Leakage may come from bolt holes (Table 10.7) or gaskets (Table 10.8). Excessive shaft end play, dynamic runout, or shaft-to-bore misalignment can cause leakage (Table 10.9). A sample of the oil should be obtained for later analysis if the seal shows signs of oil-related degradation.

10.3 LEAKAGE ANALYSIS: AFTER SEAL IS REMOVED

10.3.1 Seal Removal Procedure

Before the seal is removed, the outer case should be marked at the 12 o'clock position so the seal can be properly oriented with the housing during the failure analysis. Seal removal is not always easy, and damage may result. Care should be exercised to minimize damage that may destroy clues that will explain the reason for failure. If possible, the machinery should be disassembled and the seal pressed out of the bore using a tool that presses on the inside of the seal. Some seals have a pryout flange that can be used to remove them if the equipment cannot be disassembled. Many times, small holes are drilled in the back face of the seal. Screws are installed into the holes, and the seal is removed by pulling on the screw heads. If the seal is not severely damaged, then the seal parameters should be measured. The seal O.D. should be measured and compared to the housing bore I.D. The seal lip inner diameter should be measured and compared to the shaft diameter. The seal radial force and lip opening pressure should be measured with the proper equipment. The seal wear track width and the variation around the circumference should be recorded. The radial wall variation should be determined. Sometimes the seal is functionally tested on a bench test machine to verify leakage. The used seal's cross-sectional profile can also be determined and compared to the cross section of a new seal. The microhardness of the lip elastomer should be determined and compared to the material specifications. The last two measurements should be made after all others are complete because they are destructive.

10.3.2 Bore and Housing Inspection After Seal Is Removed

After the seal is removed, the inside of the bore should be carefully inspected and measured (Table 10.10). Spiral tool marks or axial scratches on

Table 10.7 Bolt Inspection Before Equipment Disassembly

Observation or defect	Probable cause	Corrective action
Leaks at bolt holes	1. Threads in housing tapped too deep into fluid reservoir	1. Review product machining specifications.
	2. Insufficient bolt tightening	2. Review specifications for bolt torque.
	3. Undersize bolt diameter or oversize thread tap diameter	3. Measure bolt and bolt hole for fit.
	4. Material thermal expansion incompatibility	4. Ensure that bolt and housing material have similar thermal characteristics for temperature extremes.
	5. Vibration	5. Employ locking method to ensure that bolt does not work loose.
	6. Bolt fracture	6. Check bolt loading specifica tions and operation parameters.
	7. Contamination	7. Ensure that bolt hole is free of particles or corrosive fluids prior to bolt installation.
	8. Corrosion	8. Ensure that bolt housing and bolt material are compatible with application environment.
	9. Bolt missing	9. Install specified bolt.
	10. Cross threading	10. Retap and use correct bolt.
	11. Improper bolt	11. Change to correct bolt size.
	12. Improper head type	12. Change to correct bolt.

Source: Rubber Manufacturers Association.

Table 10.8 Gasket Inspection Before Equipment Disassembly

Observation or defect	Probable cause	Corrective action
Leaks from gasketed components or joints	1. Heat aging causing stress cracking or compression set	1. Use high-temperature gasket material.
	2. Improper machining of mating surfaces	2. Review machining procedure for proper machining tech niques.
	3. Casting porosity or other hardware surface defects	3. Inspect hardware surface for visible defects prior to gasket installation.
	4. Excessive preload of gasket resulting in gasket compression set	4. Review bolt torque requirements.
	5. Gasket swelling, softening, or hardening due to chemical attack	5. Check fluid compatibility of gasket material.
	6. Torn gasket	6. Use proper installation procedures and tools.
	7. Crimped or folded gasket	7. Use proper installation procedures and tools.
	8. Gasket blowout	8. Review system pressure specifications and field application conditions. Check hardness of gasket.
	9. Dry gasket	9. Replace gasket.
	10. Wrong size	10. Use correct gasket.
	11. No sealant on gasket	11. Apply sealant.
	12. No gasket	12. Install gasket.

Source: Rubber Manufacturers Association.

Table 10.9 Shaft Inspection Before Equipment Disassembly

Observation or defect	Probable cause	Corrective action
Excessive shaft-to-bore misalignment	1. Poor initial alignment	1. Review design and assembly operations and provide accurate alignment.
	2. Seal manufactured with high radical wall variation	2. Review production quality data and adjust process.
Excessive shaft runout	1. Failed bearing; exceeded bearing load capacity; excessive wear or contamination--	1. Replace bearing.
	2. Excessive shaft deflection	2. Balance shaft and/or support shaft better.
	3. Shaft machined out of tolerance	3. Review shaft print specifications and production limits and tolerances, and adjust process.
Excessive axial end play	1. Worn thrust bearing	1. Replace bearing, check bearing load specification and gap tolerances.
	2. Shearing of lock ring or locking key	2. Check hardness of lock device and dynamic load on shaft.
	3. Loose wear sleeve on shaft	3. Check press or bond fit for sleeve.
	4. Negative stack-up in hardware tolerances	4. Review product prints.
Leakage between shaft and wear sleeve	1. Improper sleeve press fit	1. Inspect at removal.
	2. Damaged shaft under wear sleeve	2. Inspect at removal.
	3. Improperly finished shaft (chatter)	3. Inspect at removal.

Source: Rubber Manufacturers Association.

Table 10.10 Bore Inspection After Seal Is Removed

Observation or defect	Probable cause	Corrective action
Seal loose in bore	1. Undersized seal O.D.; bore size correct	1. Replace with correct seal size.
	2. Oversize bore diameter resulting from seal press fit deformation	2. Check seal to ensure proper O.D. size; increase housing radial wall in area of seal gland.
	3. Tapered bore diameter resulting from improper machining techniques	3. Specify maximum axial diameter taper.
	4. Oversize bore due to design error	4. Contact OEM for corrective action.
	5. Oversize bore not in dimensional agreement with OEM specification	5. Unit may be a rebuild; check seal O.D. diameter and order proper replacement part.
	6. Bore finish too smooth	6. Provide rougher bore or use O.D. sealant.
Seal O.D. collapsed	1. Seal O.D. too large	1. Use correct seal size.
	2. Undersize bore due to design error	2. Contact OEM for corrective action.
Spiral tool withdrawal marks on bore; leakage observed	1. Poor machining practices; tool in contact with surface during removal	1. Review machining techniques.
Scratches or gall marks on bore; leakage observed	1. Scratches and galling due to poor handling techniques	1. Review handling and shipping practices.
	2. Scratches and galling due to machining operations	2. Review machining practices.
	3. Scratches and galling to part assembly; i.e., shaft, seal, bearings	3. Review assembly practices.
Bore leakage observed	1. Leakage through imperfections and tool withdrawal marks	1. Machine and use large O.D. seal; machine and install sleeve.

Table 10.10 Continued

Observation or defect	Probable cause	Corrective action
	2. Bore surface too rough	2. Machine finish to 3.17 μm (125 μin.) or less; add O.D. sealant.
Improper lead-in chamfer	1. Chamfer lead-in not adequate to install seal due to improper chamfer angle	1. Review machining practices and product drawing.
	2. Deformation of lead-in chamfer edge due to chamfer diameter less than maximum O.D. of seal	2. Check O.D. of seal to ensure seal not oversize and check I.D. of chamfer to ensure I.D. meets drawing specifications.
	3. Chamfer not present due to machining or product drawing error	3. Review product drawing and make appropriate changes.
	4. Chamfer deformed due to seal installation	4. Increase hardness of bore material; use rubber-covered seal.
	5. Chamfer too long, causing insufficient flat area for seal retention	5. Check product drawing and chamfer angle; measure seal width to ensure proper part and fit.

Source: Rubber Manufacturers Association.

the inside of the bore will provide a leakage path. If this condition exists in the field and a new housing is not available or practical, then a field repair is necessary. An O.D. sealant can be applied to the seal O.D. and the bore I.D. to stop leakage when a new seal is installed. The bore can be carefully machined to a larger diameter to remove the marks, and a new seal with a larger O.D. can be installed. If a seal with a larger O.D. is not available, then the bore is machined and a sleeve is installed to provide the proper bore I.D. for the seal. Care must be taken to ensure that leakage will not develop between the sleeve and the housing. Leakage can sometimes result from flaws in the housing itself (Table 10.11).

10.3.3 Seal Inspection After Removal

After the seal is removed, it is inspected. Many failures occur when the seal elastomer is exposed to aggressive oils at high temperatures (Table 10.12)

Table 10.11 Housing Inspection After Seal Is Removed

Observation or defect	Probable cause	Corrective action
Porosity in housing	Casting defect	Review foundry practices.
Cracks in housing	Cracks in housing due to heat treating or mishandling	Review material heat-treating specification and handling practices.
Rough surface	Embedment of grinding media producing rough surface	Review machining practices.

Source: Rubber Manufacturers Association.

[75]. The material may become very hard, and deep cracks will develop in the lip that generate leak paths (Fig. 10.2). If the lip hardens slightly, these cracks may not be present but the lip will no longer follow shaft vibrations at high speeds, and leakage will result. This condition can be detected when a slight cracking or crazing develops when the seal lip is flexed (Fig. 10.3). Seal lip breakage can result if the material is exposed to very low temperatures with excessive dynamic runout (Fig. 10.4).

Some materials will crack when exposed to ozone. High levels of ozone are developed during welding, and short exposures of nitrile materials to these levels can be harmful. It is recommended that seals be stored away from areas where much welding takes place.

Exposure to oils at high temperatures can cause excessive softening, swelling, and eventual crumbling or reversion of silicone elastomers (Fig. 10.5). If this occurs, the silicone material should be replaced with a material that is compatible with the lubricant. Silicone materials should never be used with EP gear oils. Excessive speeds at high sump temperatures will result in excessive lip temperatures. These high lip temperatures will cause the oil to burn up and deposit a hard layer of carbon on the lip (Fig. 10.6). Both the micro and macro pumping mechanisms will be rendered useless, and gross leakage will result.

Blisters (Fig. 10.7) may also develop on the air side of the seal lip, deforming the seal lip, lifting it off the shaft, and providing a leakage gap. Blisters may shrink and even disappear when they cool. If blisters are a suspected cause of leakage, the seal should be heated and inspected while hot to see if the blisters reappear.

High lip temperatures can cause the oil film to become too thin to support the seal load. Stick-slip can develop and is characterized by a series of axial tears in the contact pattern (Fig. 10.8). The elastomer may be soft and show no evidence of hardening. Extreme cases of stick-slip can destroy the elastomeric lip completely (Fig. 10.9).

Table 10.12 Seal Inspection: Elastomer Degradation

Observation or defect	Probable cause	Corrective action
Hardened lip	1. Prolonged or excessive high-temperature exposure (Fig. 10.2)	1. Check rubber spec vs. system temperature profile.
	2. Extended dry running causing localized high temperature under lip (Fig. 10.3)	2. Check fluid level; check for shaft that is too smooth.
	3. Ozone exposure	3. If areas of rubber not at the wear track are cracked, then consider excessive solar or electrical exposure (i.e., welding near seal assembly area).
Broken lip	1. Flexing of lip at temperatures below rubber capability (Fig. 10.4)	1. Check rubber spec vs. system temperature profile; check offset, runout, and sideplay.
Softening, swelling reversions (Fig. 10.5)	1. Elastomer not compatible with lubricant	1. Check elastomer-fluid compatibility, upgrade elastomer if necessary.
	2. Exposure to solvent used during teardown	2. Review teardown procedure and elastomer compatibility with solvents.
	3. Operational contamination of fluid being sealed	3. Check for possible exposure to unspecified media coming in contact with seal.
Coked oil on lip (Fig. 10.6)	1. Excessive underlip temperature	1. Reduce seal load.
	2. Wrong lubricant for temperature requirements	2. Change lubricant (possibly a synthetic).
Blisters on lip (Fig. 10.7)	1. Excessive underlip temperature	1. Reduce seal load.
	2. Wrong elastomer for temperature requirements	2. Change elastomers.

Table 10.12 Continued

Observation or defect	Probable cause	Corrective action
Chatter marks in wear path (Fig. 10.8)	1. Stick-slip caused by high temperature and thin oil	1. Reduce seal load.
	2. Film	2. Change friction characteristics of elastomer or lubricant.
Seal lip destroyed (Fig. 10.9)	1. Stick-slip caused by high temperature and thin oil	1. Reduce seal load.
	2. Film	2. Change friction characteristics of elastomer or lubricant.

Source: Rubber Manufacturers Association.

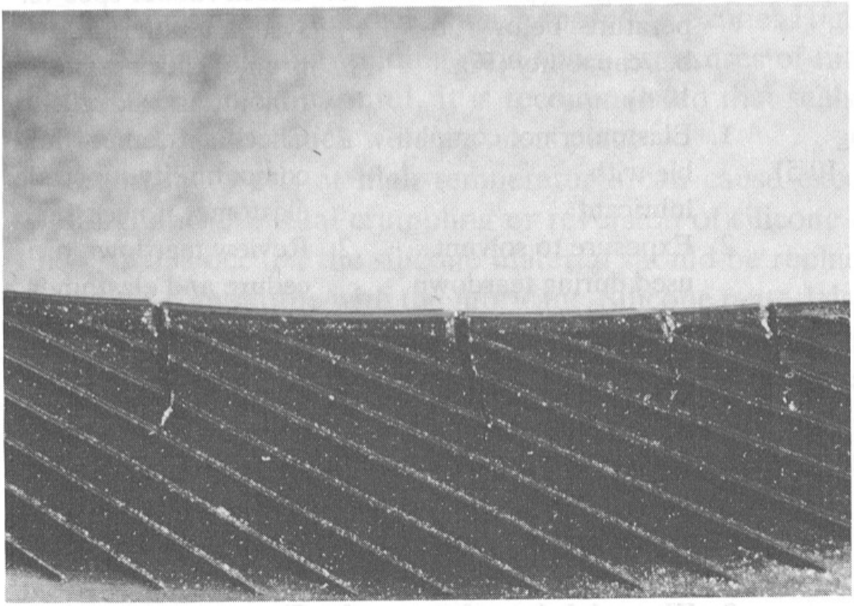


Fig. 10.2 Deep cracks in elastomer due to prolonged exposure to high tempertures.

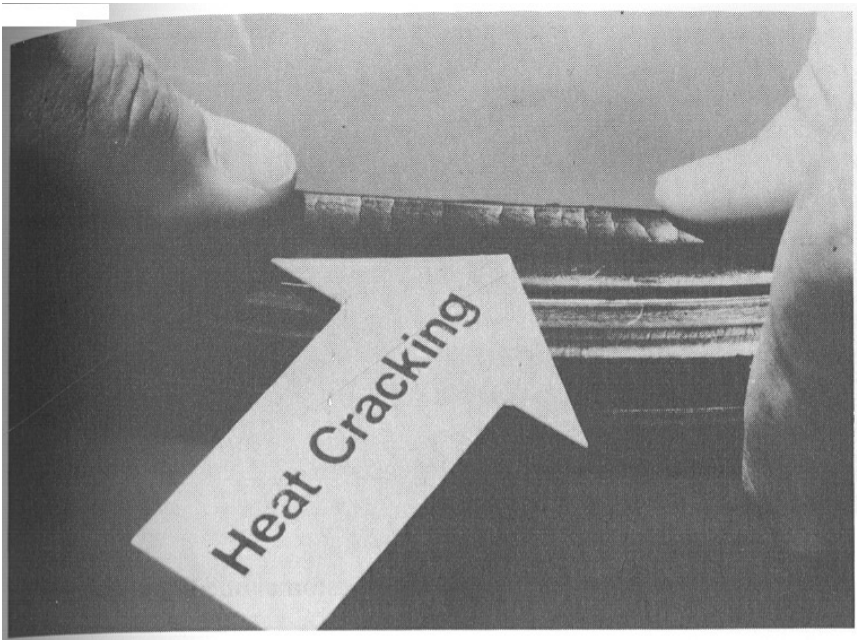


Fig. 10.3 Surface cracks due to dry running and high lip temperatures.

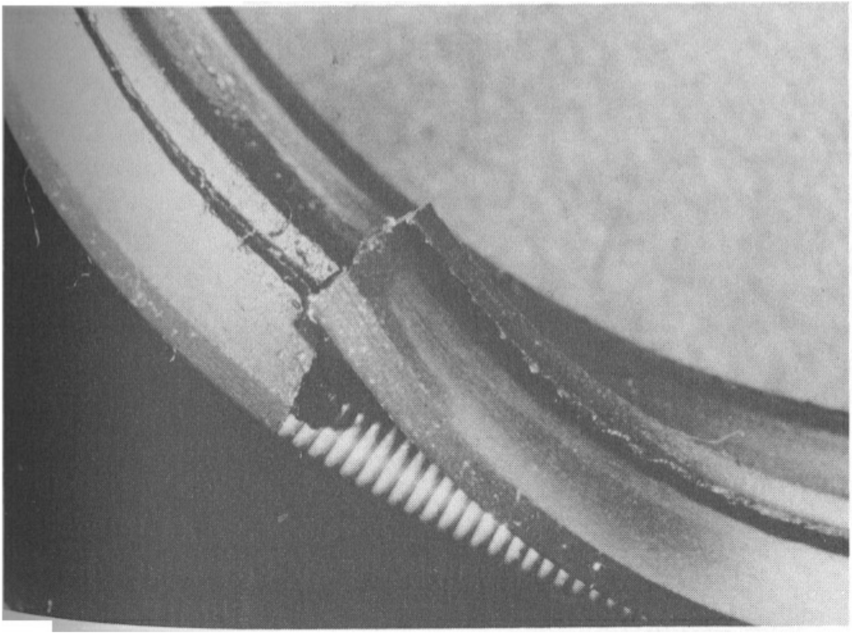


Fig. 10.4 Low application temperature and excessive dynamic runout can cause seal lip breakage.

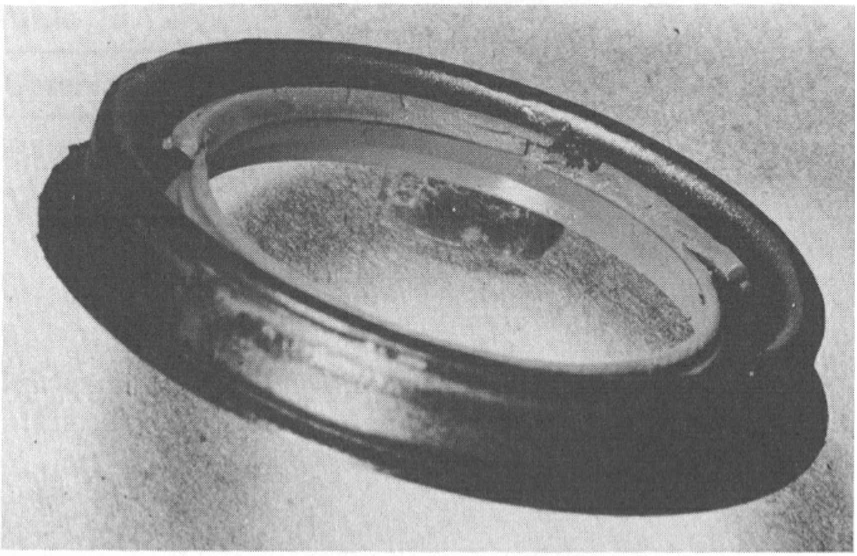


Fig. 10.5 Softening, swelling, reversion due to elastomer incompatibility with lubricant.

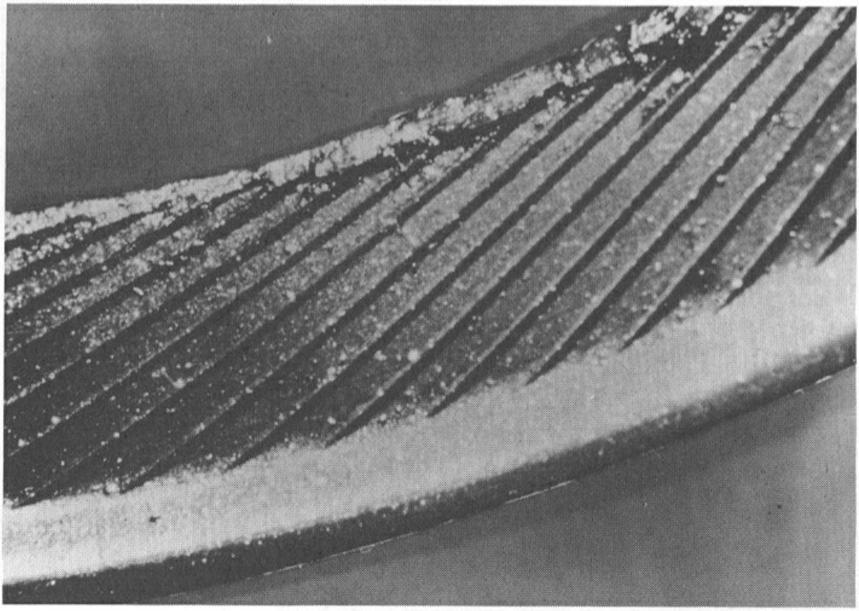


Fig. 10.6 Oil coking clogs seal lip.

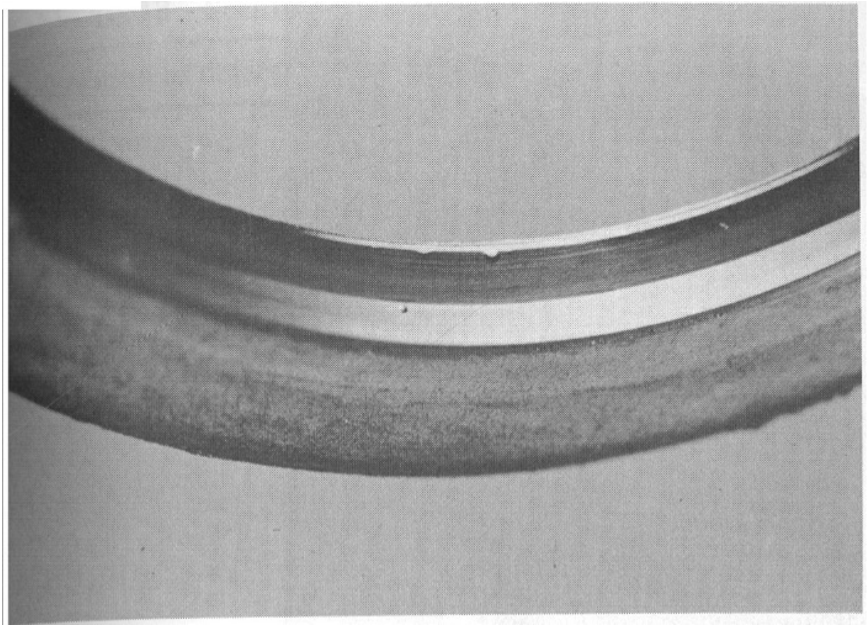


Fig. 10.7 Blisters deform seal lip.

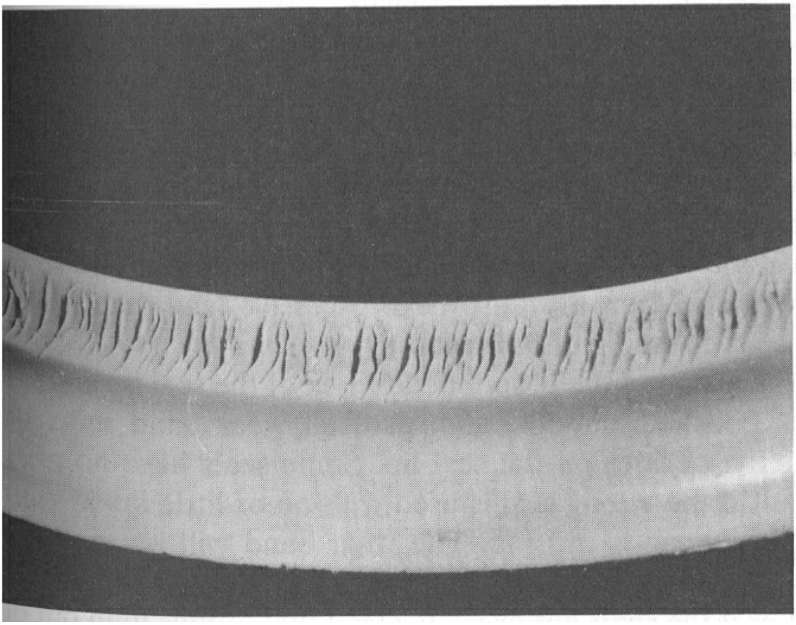


Fig. 10.8 Poor lubrication results in stick-slip and chatter marks in seal wear track.

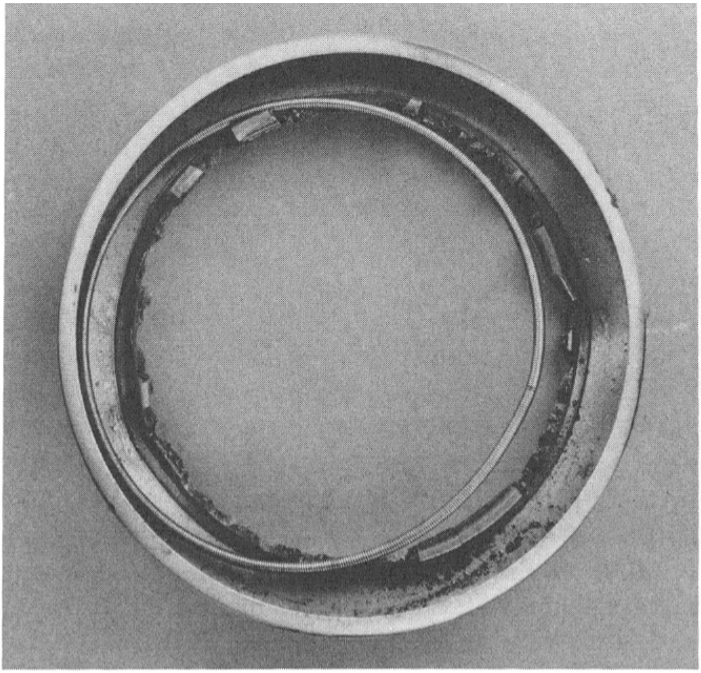


Fig. 10.9 Poor lubrication can destroy seal element.

The seal contact band is inspected to determine if an unusual wear pattern is present (Table 10.13). A wide wear track indicates the possibility of excessive pressure, excessive initial radial load, or excessive initial lip-to-shaft interference. Wide wear can also be caused by a rough shaft finish or a lack of lubrication. The shaft may also have excessive lead that pumps into the sump and starves the lip of lubrication.

Seal function depends upon the proper bedding-in of the seal lip and the shaft. The shaft is polished smooth, and an even wear band must be developed with microasperities on the seal lip. Some seals have no wear band. This can result if the wrong seal is used with no or little lip-to-shaft interference or if the spring is missing. The wear band will not develop properly if the seal is leaking at start-up. This can happen if the seal is installed backward or if the shaft has excessive lead that pumps fluid out of the sump. Improperly formulated elastomers may also leak early, and a wear band will not develop on the lip. If the seal is rotating, excessive speeds may cause the lip to lift off the shaft, which will result in early leakage.

Table 10.13 Seal Inspection; Contact Band Wear Patterns

Observation or defect	Probable cause	Corrective action
No lip wear	1. Little or no interference with shaft	1. Check for missing spring; check shaft size. Check for seal distortion; check to be sure correct seal is used.
	2. Seal installed backward	2. Correct installation procedures.
	3. Heavy leakage from start-up	3. Check shaft for lead, check to be sure shaft rotation is in right direction if uni directional (helix) seal is used. Check lip for defects. Check shaft and seal size.
	4. Dynamic lift off (centrifugal force)	4. Check spring to be sure load is correct; use proper spring for application.
Excessive wear	1. Excessive lip load	1. Check system pressures. Check to see if vents are clogged. Check seal-to-shaft interference to be sure it is not excessive. Check spring load; use proper spring.
	2. Rough shaft finish	2. Measure shaft finish; use proper roughness.
	3. Insufficient lubrication at seal lip	3. Provide lubrication on air side of seal or between lips.
Eccentric wear (Fig. 10.10)	1. Seal cocked	1. Use correct assembly tools and procedures.
	2. Excessive STBM	2. Check shaft-to-bore offset; correct hardware problems.
	3. Cocked shaft	3. Check for bent shaft or worn bearings; replace components if necessary.

Table 10.13 Seal Inspection; Contact Band Wear Patterns

Observation or defect	Probable cause	Corrective action
	4. Excessive RWV	4. Measure seal RWV and compare to wear pattern. Notify supplier for quality checks if RWV is excessive.
	5. Side load applied to shaft	5. Check for worn bearings; replace if necessary.

RWV = radial wall variation.

Source: Rubber Manufacturers Association.

The wear band may have a region of heavy or wide wear with a region of light wear about 180° away. This is caused by seal or equipment misalignment. The seal may be cocked in the bore, there may be excessive shaft-to-bore misalignment, or the shaft may be cocked (Fig. 10.10). The seal may also have excessive radial wall variation that causes eccentric wear. Radial wall variation results when the center of the seal lip I.D. is not concentric with the center of the seal O.D.

The seal lip may also be damaged by burrs on the shaft or by sharp edges or burrs on installation tools. Damage can also result if the seal is installed over keyways or splines without proper installation tools (Table 10.14).

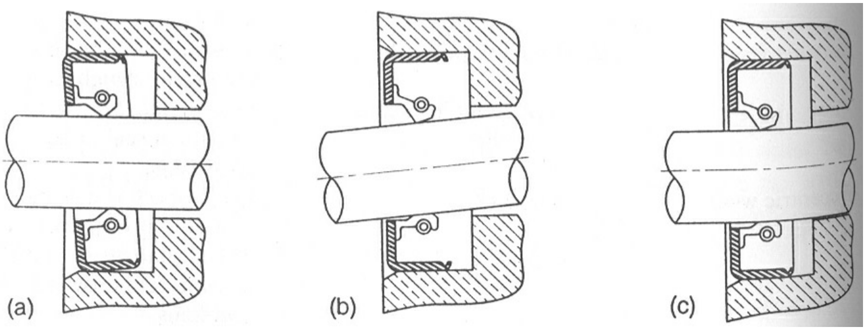


Fig. 10.10 Eccentric lip wear is caused by seal or equipment misalignment. (a) Cocked seal. (b) Cocked shaft. (c) Excessive shaft-to-bore misalignment.

Table 10.14 Seal Inspection: Lip Damage, Lip Tears, and Lip Turn-under

Observation or defect	Probable cause	Corrective action
Nicks on seal contact point	1. Sharp edge or burrs on end of shaft	1. Inspect shaft condition for burrs or sharpness; remove burrs or provide installation bullet.
	2. Sharp edge or burrs on installation tool	2. Check installation device for sharp edges or burrs; repair tool or use soft plastic sleeves or bullets.
	3. Seal installed over keyway or splines	3. Provide installation sleeve for splines or keyways.
	4. Trimming knife cuts	4. Check supplier's knife trimming methods.
	5. Nibbled appearance at sealing edge	5. Defects may be caused by bulk finishing or handling by supplier.
	6. Cuts from packaging methods	6. Check supplier's packaging and shipping methods.
Tear in seal lip section	1. Stress fatigue in flex section	1. Check system pressure; seal may be deformed in I.D. flex section; clean vents; use pressure seal design.
	2. Bond separation at I.D. of metal case	2. Check seal for bond.
	3. Migration of low-temperature crack	3. Check lip contact area for minor cold cracks; suspect severe side load or excessive runout at low temperature.
	4. Circumferential tear behind lip	4. Look behind lip at base for circumferential tear caused by pressure or fatigue.
	5. Tears caused during disassembly or removal	5. Review teardown and seal removal methods and check tools used.

Table 10.14 Continued

Observation or defect	Probable cause	Corrective action
Lip section inversion	1. Oil to air-side shaft assembly	1. Provide installation aid such as bullet nose for shaft.
	2. Lack of proper concentricity assembly	2. Provide centering aid for assembly such as locating pins.
	3. Improper shaft chamfer	3. Use assembly bullet; provide proper shaft chamfer.

Source: Rubber Manufacturers Association.

Tears in the flex section are usually caused by fatigue and are the result of excessive system pressure. Vents should be cleaned to reduce pressure, or a different seal design should be used if the system has no vents.

If poor installation practices are followed, the seal lip may be turned under during installation, spring popout may occur, and gross leakage will result. Lip inversion can be caused by an improper chamfer on the shaft. Assembly sleeves should be used, and the shaft manufacturer should be contacted to provide a chamfer that meets recommended specifications.

The seal O.D. should also be measured and examined (Table 10.15). If the seal O.D. is too small or too large for the application bore, then the wrong seal may have been selected. Many times, the used bore is larger than a new one. This may happen if the bore has been remachined or if the housing walls are too thin and the bore expanded when the previous seal was installed. Sleeves may be required to correct this problem in the field, or a seal with a larger O.D. can be used. The housing manufacturer should be contacted, and a redesign with thicker bore walls may be required to prevent deformation when the seal is installed. If a metal O.D. has severe axial scratches, then a damaged bore or chamfer may be the culprit. If no damage is found, then the scratches may be generated during the metal forming operation. The seal manufacturer should be contacted for corrective action.

Many seals include a garter spring, and an improper spring design, wrong dimensions, missing spring, corroded spring, separated spring, or double spring may cause seal leakage (Table 10.16). If any of these conditions is found, the seal supplier should be contacted for corrective action.

Table 10.15 Seal Inspection: Seal O.D.

Observation or defect	Probable cause	Corrective action
Seal O.D. too large	1. Wrong size for application	1. Use correct seal size.
Seal O.D. too small	1. Wrong seal for application	1. Use correct seal size.
	2. Bore or housing reworked	2. Check housing print; also check for evidence of rework such as chuck marks.
	3. Bore walls deformed during previous seal installation	3. Provide thicker housing walls.
Severe scratches on metal seal O.D.	1. Damaged bore or bore chamfer	1. Check condition of bore and chamfer; repair as required.
	2. Die scratches from case forming operation	2. Check under O.D. coating or sealant for scratches; contact seal supplier.
Rubber O.D. nonfills or cuts	1. Improper seal manufacture; prep weight or shape out of spec	1. Consult seal manufacturer.
	2. Bore chamfer sharp or damaged	2. Inspect bore chamfer; repair as necessary; contact bore supplier for proper chamfer.
Rubber O.D. peeled	1. Poor rubber bond to case	1. Consult seal manufacturer.
	2. Lack of lubrication of O.D. at assembly	2. Lube seal O.D. before installation.
	3. Lack of proper lead-in chamfer	3. Check bore chamfer condition; repair as necessary; contact bore supplier for proper chamfer.
Rubber O.D. hardened	1. Excessive heat in application	1. Check for other evidence of overheating; change seal material.
	2. Improper seal material selection	2. Check materials vs. operating temperature or lube compatibility; change seal material.

Source: Rubber Manufacturers Association.

Table 10.16 Seal Inspection: Garter Spring

Observation or defect	Probable cause	Corrective action
Missing spring	1. Seal may never have had a spring.	1. Check for spring coil marks in spring groove, also light wear on primary lip. Consult with seal supplier.
	2. Spring may have become dislodged during seal or shaft installation.	2. Check installation procedures. Consult with seal supplier.
	3. Spring joint may have separated due to improper spring nib configuration or improper seal installation.	3. Check installation procedures; check garter spring nib if spring is available. Consult with seal supplier.
Corroded spring	1. Spring may not have had proper rust preventive applied by supplier.	1. Check new seals from same supplier. Consult supplier.
	2. Application may be exposed to excessive moisture.	2. Check for moisture in lube or corroded components; specify stainless steel spring.
	3. Application may contain a corrosive fluid.	3. Specify stainless steel spring.
	4. Seal may have been improperly packaged and/or stored prior to installation.	4. Check service stock; consult supplier.
	5. Wrong spring material.	5. Consult supplier.
Spring does not meet specifications	1. Wrong spring on seal (excessive wear or no wear on primarily lip).	1. Check seal drawing for spring dimensions. Consult supplier.
	2. Spring not properly . normalized	2. Check for proper heat treatment; check seal drawing. Consult supplier.
	3. Improperly manufactured spring.	3. Check seal drawing for spring dimensions. Consult supplier.
More than one spring in groove	1. Malfunctioning spring installation.	1. System audit at supplier. Consult supplier.
	2. Loose springs at seal installation station; extra spring installed by assembler.	2. Review installation station; remove any loose springs; review seal design and packaging. Consult supplier.

Source: Rubber Manufacturers Association.

10.3.4 Shaft Inspection After Seal Is Removed

The shaft should be inspected to determine if it had a role in the sealing system failure (Table 10.17). If the shaft is too large, lip inversion may result or excessive radial load will cause overheating and premature failure. If the shaft is too small, there will not be enough interference to follow shaft vibrations and dynamic runout; early leakage will result.

A shaft chamfer that is not within recommended specifications may cause lip inversion or spring popout during installation. Burrs or sharp edges on the chamfer may damage the seal lip during installation. Excessive shaft roughness will accelerate lip wear and lead to early leakage. A shaft surface that is too smooth may result in a poorly lubricated lip, and stick-slip may result. Machine lead may screw oil out of the sump and cause instant leakage. Machine lead can also screw oil into the sump, which may

Table 10.17 Shaft Inspection After Seal Is Removed

Observation or defect	Probable cause	Corrective action
Shaft too large or too small	1. Improper shaft manufacturing techniques	1. If oversize, machine and grind to proper diameter. Consult shaft supplier.
Shaft chamfer not within specification	1. Improper shaft manufacturing technique	1. Consult shaft supplier, Use installation bullet or sleeve.
Sharp edges or burrs on shaft edge	1. Improper shaft manufacturing technique	1. Consult shaft supplier. Use installation bullet or sleeve.
Shaft roughness not within specification	1. Improper finishing techniques	1. Use a thin-wall wear sleeve; consult shaft supplier; use recommended finish specs.
Shaft too soft (less than Rc 30)	1. Improper or no hardening	1. Harden shaft to Rc 45; use wear sleeve; consult shaft supplier.
Excessive machine lead on shaft	1. Improper finishing technique	1. Consult shaft supplier; use plunge grinding; install wear sleeve with proper finish.
Wrong shaft material	1. Improper material selection	1. Consult shaft supplier; select material that is compatible with application needs; use wear sleeve.

Table 10.17 Shaft Inspection After Seal Is Removed

Observation or defect	Probable cause	Corrective action
Leakage between wear sleeve and shaft	1. Wear sleeve too loose	1. Use sealant between sleeve; use proper sleeve and shaft specifications.
	2. Wear sleeve damaged during installation	2. Check chamfer for proper dimensions and burrs.
Deformed sleeve	1. Sleeve I.D. too small	1. Check dimension; consult wear sleeve supplier.
	2. Shaft O.D. too large	2. Check dimension; consult shaft supplier; machine shaft to smaller diameter.
Scratches or nicks across seal contact area	1. Damage during machining and assembly	1. Check handling procedures; use nylon mesh sleeves to protect shaft surface.
	2. Shaft too soft	2. Harden shaft to Rc 45.
Shaft corrosion	1. Wrong material for application	1. Use corrosion-resistant shaft material; use corrosion-resistant wear sleeve.
	2. Improper storage techniques	2. Oil or grease shafts before storage.
	3. Contaminant ingestion	3. Change seal design to exclude contaminants.
Shaft discoloration in seal contact region	1. Excessive temperature	1. Check interference levels, lubricant, shaft smoothness, bearing preload; use cooling coils to reduce heat; add lubrication, change seal.
Coked oil buildup in wear band	1. Temperature exceeding oil capabilities	1. Change oil; reduce operating temperature with cooling coils.

Source: Rubber Manufacturers Association.

cause poor lip lubrication, stick-slip, and catastrophic failure. If the shaft is too soft, it may be damaged during handling, causing instant leakage. An improper selection of shaft material may result in excessive corrosion, poor heat transfer, and excessive shaft wear or grooving. If any of these conditions exists, the shaft supplier should be contacted to develop a permanent solution.

Properly finished wear sleeves can be used to provide a temporary solution. The existing shaft may require machining to a smaller diameter, or thin-walled wear sleeves may be used. Care must be taken to ensure that the wear sleeve fits properly. Leakage may seep through gaps between the sleeve and the shaft. If the sleeve is too loose, then the sleeve may not rotate with the shaft and rubbing of the two members will generate excessive heat. If the sleeve is too tight, distortion may result during installation.

Scratches or nicks in the seal contact region may cause leakage. The shaft should be protected during handling with nylon mesh sleeves or their equivalent. The shaft should be hardened to Rc 45.

Corroded shafts may result from improper storage techniques. The shafts should be oiled or greased before storage to prevent corrosion. If corrosion is observed after installation, the shaft material may not be compatible with the lubricants or outside contaminants. A corrosion-resistant material can be used for the shaft, or a seal design that minimizes the ingestion of outside contaminants can be employed.

Excessive temperatures in the seal area can cause the shaft to discolor and the oil to break down. Sludge can build up in the seal contact area, which will destroy seal function and cause leakage. Cooling coils are often installed in the machinery to reduce sump temperatures, protect the oil, and prevent sludge buildup.

The shaft surface should be inspected to determine where the seal lip contacted the shaft. The wear track location and its condition can help the investigator determine the cause of leakage (Table 10.18) [137]. The position of the wear track can tell the investigator if the seal was properly installed (Fig. 10.11). Installation tools with proper positive stops on the shaft end or the housing bore will ensure that the seal is positioned properly without cocking.

Sometimes pressure or springback (prevalent with rubber O.D. seals) will cause the seal to change position. A snap ring or backup washer will prevent seal popout. Lubricating the O.D. of a rubber seal will minimize popout.

The wear track on the shaft may be wider than the seal contact width. This indicates that there is excessive axial motion or end play. The bearings may need replacement and may be the root cause of observed leakage. Seal and/or shaft cocking can also result in this wear condition. Sometimes the

Table 10.18 Shaft Wear Track Inspection After Seal Is Removed

Observation or defect	Probable cause	Corrective action
Lip wear band on wrong position on shaft; may ride on chamfer or contact internal machine components	1. Seal installed backward	1. Install seal correctly.
	2. Seal not seated properly (Fig. 10.11a, b)	2. Review installation procedures; use tools with positive stops to properly locate the seal.
	3. Seal moves after installation	3. Check installation procedure; snap ring may be required if pressure is a problem.
Shaft wear band wider than seal wear band	1. Excessive axial motion or end play	1. Check bearings and replace if defective or worn.
	2. Seal cocking	2. Correct seal assembly procedures.
	3. Shaft cocking	3. Check bearings and equipment alignment; correct as required.
Excessive shaft wear and grooving	1. Shaft too soft	1. Harden shaft to Rc 45; use hardened wear sleeve.
	2. Contaminant ingestion	2. Use seal design that minimizes contaminant ingestion.
	3. Contaminant in lubricant	3. Change filters frequently; change oil.
	4. Too much lip interference	4. Check specifications; reduce lip interference if required.
	5. Excessive eccentricity	5. Check bearings and equipment alignment; replace bearings if required.

Source: Rubber Manufacturers Association.

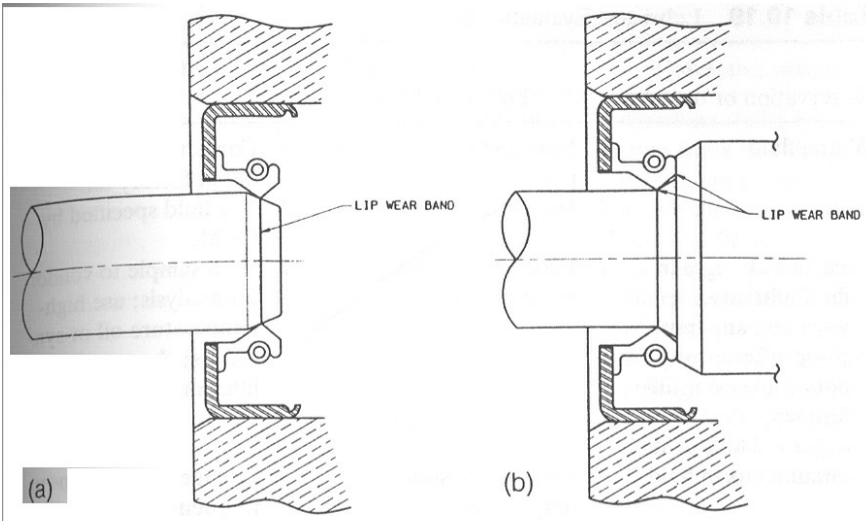


Fig. 10.11 (a) Shaft not located properly and (b) seal installed too deep in bore contacts internal components. (Courtesy of Rubber Manufacturing Association).

shaft is deeply grooved; this can often be corrected by hardening the shaft to Rc 45. External contaminants or particles in the oil can also create deep grooves in the shaft. More frequent oil and filter changes will reduce the concentration of particles in the oil that cause shaft grooving. A seal design to exclude contaminants will prevent ingress of dust, mud, and water, which also contribute to shaft grooving.

10.4 LEAKAGE ANALYSIS: LUBRICANT EVALUATION

Inspection of the oil can also reveal clues to the root cause of sealing system failures (Table 10.19). The wrong fluid can attack elastomers and cause early failures. Equipment users should always use the lubricant recommended by the OEM when they replace oils. The lubricants can degrade when exposed to excessive temperatures.

When lubricants break down, the viscosity increases dramatically, lubricity is lost, sludge builds up, and leakage results. Degraded oil will often smell and change color. Sump coolers will reduce oil degradation, but a high-temperature synthetic lubricant may be necessary.

Contaminants in the fluid will contribute to shaft grooving and early leakage. The contaminants may be sludge particles from degraded oil, or

Table 10.19 Lubricant Evaluation

Observation or defect	Probable cause	Corrective action
Wrong fluid	1. Incorrect factory fill procedures	1. Correct procedures for initial factory fill.
	2. Wrong fluid fill in field	2. Use fluid specified by OEM.
Excessive changes in fluid lubricity, viscosity, apparent color differences, noticeable odor difference, degraded lubricant	1. Excessive system temperatures	1. Send sample to vendor for analysis; use high-temperature oil or synthetics; change oil and filter more frequently.
Contaminants in fluid	1. Excessive system temperatures	1. Change oil filters more frequently.
	2. Contaminants in initial fill	2. Check with oil supplier.
	3. Ingress of contaminants	3. Change seal design to exclude contaminants.
	4. Inadequate cleaning of unit before assembly	4. Review cleaning procedures.
	5. Wear debris of bearings shafts or other components	5. Change bearings, flush out unit, check other components for excessive wear.

Source: Rubber Manufacturers Association.

they may result from poor cleaning of the unit before assembly. Sometimes purchased oil will have contaminants because of poor practices at the refinery. Wear debris from bearings, shafts, and other internal components can also contaminate the oil. Frequent oil and filter changes will help minimize shaft and seal lip damage. The most common source of oil contamination is the external environment. A seal design that maximizes contaminant exclusion should be used to reduce contamination from this source.

10.5 SUMMARY

The symptoms, possible causes for them, and corrective actions are summarized in Table 10.20 for easy reference.

Table 10.20 Summary of Seal System Failure Analysis

Observation or defect	Probable cause	Corrective action
Early lip leakage	1. Nicks, tears, or cuts in seal lip	1. Examine shaft; eliminate burrs and sharp edges; use correct mounting tools to protect seal lip from splines, keyways, or sharp shoulders; handle seals with care; keep seals packaged in storage and in transit.
	2. Rough shaft	2. Finish shaft to 10--20 μin. rms or smoother.
	3. Scratches or nicks on surface	3. Protect shaft after finishing.
	4. Lead on shaft	4. Plunge-grind shaft surface.
	5. Excessive shaft whip or runout	5. Locate seal close to bearings; ensure good, accurate machining practices.
	6. Cocked seal	6. Use correct mounting tools and procedures.
	7. Paint on shaft or seal element	7. Mask seal and adjacent shaft before painting.
	8. Turned-under lip	8. Check shaft chamfer for roughness; check shaft chamfer for steepness; use recommended lead chamfer; use correct mounting tools and procedures.
	9. Damaged or "popped out" spring	9. Use correct mounting tools and procedures to apply press-fit force uniformly; protect seals in storage and transit.
	10. Damaged or distorted case	10. Use correct mounting tools and procedures to apply press-fit force uniformly; protect seals in storage and transit.

Table 10.20 Continued

Observation or defect	Probable cause	Corrective action
	11. O.D. sealant on shaft or lip element	11. Use care in applying O.D. sealant; purchase precoated seals.
Lip leakage, intermediate life	1. Excessive lip wear	1. Check seal cavity for excessive pressure; provide vents to reduce pressure; provide proper lubricating for seal; check shaft finish; make sure finish is 10--20 µin.
	2. Element hardening and cracking	2. Reduce sump temperature if possible; upgrade seal material; provide proper lubrication for seal; change oil frequently; change seal design.
	3. Element corrosion and reversion	3. Check material-lubricant compatibility; change material.
	4. Excessive shaft wear	4. Check shaft hardness; harden to Rockwell C30 minimum; change oil frequently to remove contaminants; use dust lip in dirty atmosphere.
O.D. leakage	1. Scored seal O.D.	1. Check housing machining; use 125 µin. rms; check edges on housing bore; use recommended chamfer; remove burrs.
	2. Damaged seal case	2. Use correct mounting tools and procedures to apply press-fit uniformly; protect seals in storage and transit.

Source: CR Industries.

11

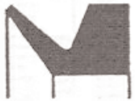
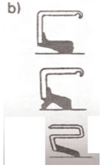
Radial Lip Seal Applications

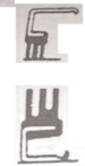
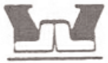
11.1 STANDARD SEALS

Many seal manufacturers offer a catalog of standard seal designs that can be used for a variety of applications. Standard seals are preferred by many seal users who have low-volume applications and cannot afford to pay for the tooling necessary to make a custom seal design. Typical seal designs that may be found in manufacturers' catalogs are described in Tables 11.1--11.3.

Non-spring-loaded seals (Table 11.1) are used to exclude contaminants and retain grease. The all-rubber V-ring (type a) offers the maximum protection against external contaminants [118]. It is hand-fitted by stretching the seal over the shaft. The axial lip will run against a bearing race, the heel of an oil seal, or the housing of the application. The V-ring is often used in tandem with a spring-loaded seal that is used to retain lubricants. Non-spring-loaded single-lip seals (Table 11.1, type b) are used to retain grease when the lip faces the bearing. The same seal can become an excluder if the lip faces the outside environment. The dual-lip version can contain the grease and exclude light contaminants. The seal configuration can be strengthened if a convoluted metal case is employed. These designs can function at up to a maximum of 3 psi and a maximum shaft surface speed of 2000 fpm. Multilip non-spring-loaded seals (Table 11.1, type c) are used for excluding heavy contamination at low speeds (500 fpm). A typical

Table 11.1 Typical Non-Spring-Loaded Standard Seal Configurations

Seal type	Description	Lip type[a]	Temp. range (°C)	Max. shaft DRO[b]	Maximum STBM[C]	Maximum pressure	Maximum shaft surface speed
a) 	All rubber V-ring for maximum contaminant exclusion; axial end play must not exceed lip length.	N F (static) F (dyn)	--40 to 100 --46 to 232 --23 to 111	Not applicable	Not applicable	10 psi at 0--3000 fpm	3000 fpm
b) 	Non-spring-loaded single-lip seals for grease retention or contaminant exclusion at low speeds.	P N S	--40 to 149 --54 to 107 -73 to 163	-0.003 in. TIR at 0--2000 rpm	0.005 in. at 0--2000 fpm	3 psi at 0--2000 fpm	2000 fpm

c) 	Multilip seals for excluding heavy contaminants; can seal on the shaft or on the inside of the hub (external seal).	N	--54 to 107	0.003 in. TIR at 0--2500 rpm	0.010 in. at 0--500 fpm	3 psi at 0--500 fpm	500 fpm
d) 	Metal face seal to exclude heavy contaminants and retain lubricants.	N	--54 to 107	Not applicable	Not applicable	Intermittent to 50 psi, continuous to 25 psi	Intermittent to 700 fpm, continuous to 350 fpm

[a] N, nitrile; P, polyacrylate; S, silicone; F, fluoroelastomer.
[b] DRO = dynamic runout.
[c] STBM = shaft-to-bore misalignment.
Source: Chicago Rawhide.

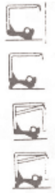
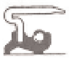
application would be a disk harrow used in farming. If heavy contaminants are present and lubricants must be retained at low pressure (25 psi) and low speed (350 fpm), then metal face seals are often used. A typical application would be sealing the track rollers that drive a tank or heavy construction vehicles. The metal face seals have two faces with lapped surfaces that are loaded by a rubber component. One face is stationary, and the other rotates. The design shown as type d in Table 11.1 is loaded by two rubber Bellville washers, but O-rings are often used instead.

Spring-loaded seals are typically used to contain fluids (Table 11.2, type a). A secondary radial lip will exclude light contaminants, and an inner cup will improve seal case rigidity, protect the lip, and help retain the spring. The maximum shaft speed is typically 3600 fpm (19.6 m/sec), and the maximum pressure is 10 psi. Seal designs with heavy lip sections will extend the maximum pressure to 90 psi, but the maximum speed will be reduced to 2000 fpm (10.9 m/sec) (Table 11.2, type b). Metal-clad seals are typically more economical and more available than rubber O.D. seals. One mold for a given shaft size can be used to make many different combinations of seal case outer diameter and width. Each combination of shaft size, seal O.D., and width requires a unique mold if the seal has a rubber O.D. Metal O.D. seals are often coated with a sealant that helps prevent bore leaks.

Seals for shaft diameters larger than 8 in. (203 mm) are usually referred to as large-diameter seals, and special designs are required (Table 11.3). They are available in sizes up to 180 in. (4600 mm). The volume demand for such seals is low, and special manufacturing techniques are employed. Large-diameter all-rubber V-rings (Table 11.3, a) can be cut and revulcanized or spliced to the proper size. A spring or clamp can be installed to prevent expansion of the seal due to centrifugal force at high speeds. Special seal designs (Table 11.3, b) have a stainless steel clamp and an exial lip that seals against the end of a large rotating shaft to exclude water and other contaminants.

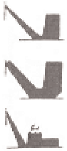
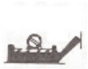
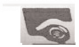
All-rubber shaft seals with springs are used to contain fluids (type c, Table 11.2). The seal can be split to make it easy to install. The seal is installed in the groove with the split at the top. The spring is joined with a hook-and-eye configuration or with a threaded connector. The size of the seal can be adjusted by cutting a solid packing, removing a portion, and splicing the ends together to give a smaller seal. If a larger seal is desired, a solid packing can be cut and a piece from a second packing spliced onto the first packing to create a larger seal. The all-rubber packings can be rolled into a steel case to provide a rigid metal-clad seal (type d, Table 11.3). The large metal case is usually formed during a spinning operation. Various combinations of lip configurations can then be developed without large investments in molds or stamping dies. Large-diameter seals are used in

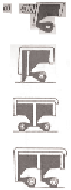
Table 11.2 Typical Spring-Loaded Standard Seal Configurations

Seal type	Description	Lip type[a]	Temp. range (°C)	Max. shaft DRO[b]	Maximum STBM[c]	Maximum pressure	Maximum shaft surface speed
a) 	Spring-loaded seals to retain lubricants. Secondary radial lip excludes light contaminants. Inner cup adds additional strength.	P N S F	--40 to 149 --54 to 107 --73 to 163 --40 to 204	.020 in. TIR at 0--1000 rpm; 0.015 in. TIR at 1000--2500 rpm; 0.010 in. TIR at 2500--4500 rpm	0.015 in. at 0--1000 fpm; 0.010 in. at 1000--3600 fpm	10 psi at 0--1000 fpm; 5 psi at 1000--2000 fpm; 0 psi at 2000--3600 fpm	3600 fpm
b) 	Heavy duty elastomeric lip for pressure with rotating shaft.	N F	--54 to 107 --40 to 149	0.005 in. TIR at 0--2000 rpm	0.005 in. at 0--2000 fpm	90 psi at 0 fpm; 50 psi at 1000 fpm	2000 fpm

N, nitrile; P, polyacrylate; S, silicone; F, fluoroelastomer.
DRO = dynamic runout.
STBM = shaft-to-bore misalignment.
Source: Chicago Rawhide.

Table 11.3 Typical Large-Diameter Seal Construction

Seal type	Description	Lip type[a]	Temp. range (°C)	Pressure tolerance	Maximum STBM[b]	Maximum DRO[c]	Maximum shaft surface speed
a) 	V-rings for contaminant exclusion. Can be clamped with a spring for high-speed applications.	N F	--40 to 100 -23 to 149	Poor	Over 7.875 in. max; STBM is 0.157 in.		Support at fpm required. None: 1--1600 fpm. Axial: 1600--2000 fpm Axial and radial: 2000--3000 fpm.
b) 	Axial clamp excludes water.	N	--54 to 107	Poor	Good	Good	5000
c) 	All rubber, spring-loaded. May be solid or split.	N F	--54 to 107-- 40 to 204	Fair	0.062 in.	0.093 in. TIR	3000

	Metal-clad seals. Many design combinations are available because of construction techniques.	N F	--54 to 107 --40 to 204	10--15 psi	0.062 in.	0.093 in. TIR	5000

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[a] N, nitrile; F, fluoroelastomer.

[b] STBM = shaft-to-bore misalignment.

[C] DRO = dynamic runout.

Source: Chicago Rawhide.

paper mills, rolling mills, dragline mining operations, propeller shafts for ships, and many other applications.

11.2 DIESEL ENGINE CRANKSHAFT

One of the most difficult radial lip seal applications is the diesel engine rear crankshaft. The crankshaft speeds can be high, and the shaft diameter is relatively large, which results in a high shaft surface speed. The seal is located in a position that splashes oil directly at the lip, but the application has sparse lubrication (Fig. 11.1), which results in high lip temperatures. Piston slap can cause large random shaft deflections that make it difficult for the seal lip to follow. Stick-slip, or lip chatter, oil degradation, and oil coking at the lip may occur and result in leakage. Diesel engine oils contain additives that can attack the elastomer, causing degradation that results in leakage. Leakage in this application is undesirable because an expensive engine and/or transmission teardown is required to replace the seal. The equipment may also be out of service for some time.

Early seal designs were conventional radial lip seals with the sealing element made from either a silicone or fluoroelastomer material (Fig. 11.2a). When lubricating oils became more complex with various additive packages, the seal life deteriorated. Silicone materials became soft and

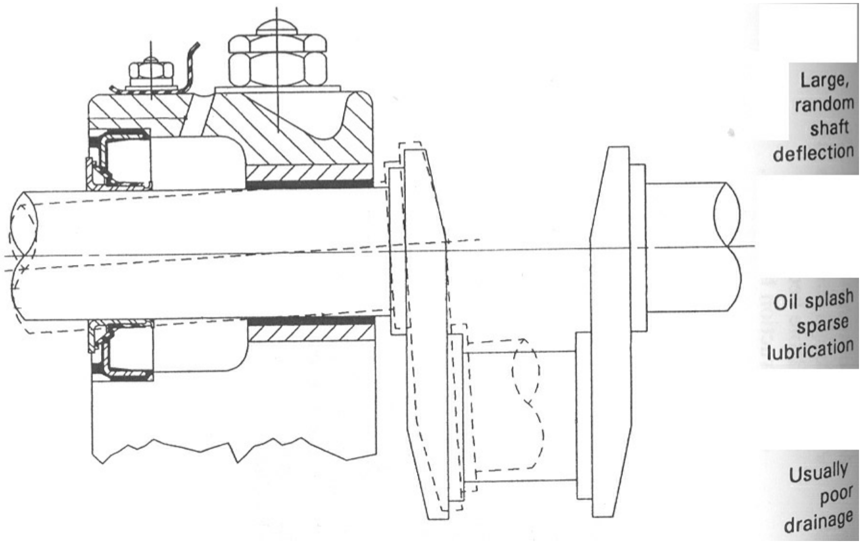


Fig. 11.1 Diesel engine crankshaft application. (Courtesy of Chicago Rawhide.)

disintegrated, which resulted in gross early leakage. Fluoroelastomer seals resisted chemical attack, but the lubricating properties of the new oils with fluoroelastomer were less than those of the old oils. The poor lubrication resulted in lip chatter or stick-slip, which generated high temperatures and damaged the seal, and early leakage occurred. High lip temperatures caused the oil to burn, sludge would build up on the lip, and leakage resulted. A PTFE wafer was bonded to a conventional elastomeric lip (Fig. 11.2b) to form a laminated seal [25]. This was done to eliminate stick-slip. A helix pattern was molded into the PTFE to provide a positive pumping action to improve seal reliability. The PTFE portion of the lip was easily damaged, and this design proved to be unreliable.

Many of the problems associated with the elastomeric lip seal were resolved by replacing the entire elastomer lip with a PTFE lip [29, 225]. The early designs were assembled and clinched between stampings. A gasket prevented oil from leaking through the seal between the PTFE wafers and the metal casing. Some seal designs have a single lip, and others use dual opposed lips to seal the oil and exclude contaminants (Fig. 11.2c). PTFE is a stiff material that cannot develop microasperities similar to those found in an elastomeric seal lip. As a result, the PTFE does not develop a natural pumping ability like a typical well-formulated elastomer. A spiral groove is machined into the surface of the primary lip that screws oil back into the sump. The pumping elements can also be coined into the PTFE wafer. The spiral grooves act as a screw thread, and the PTFE seal will function only if the shaft always rotates in the same direction. Instant leakage will occur if the direction of shaft rotation is reversed. The direction of rotation for engine crankshafts is constant, and reversal never occurs. The unidirectional spiral hydrodynamic aid used with PTFE wafers is ideally suited for this application. The PTFE lips can be easily damaged during handling or installation. Seals are often shipped on a wear sleeve, and the seal and wear sleeve are installed simultaneously to minimize lip damage. If the application does not require a wear sleeve, then a throw-away plastic shipping sleeve is used to protect the seal.

The seal manufacturer must assemble all components with high precision and use a rubber gasket to prevent internal leakage. If the components are not clamped tightly enough, the wafers may slip and gross leakage will result. These problems are eliminated with the design shown in Fig. 11.2d. The PTFE wafer is machined from the billet in a conical shape, treated by chemically etching the PTFE surface with a sodium metal slurry, and cement is applied. The wafer is then bonded to rubber, which is bonded to the metal case. A hydrodynamic spiral pattern is also coined into the wafer during the molding operation to provide a positive pumping action during shaft rotation. The single conical wafer acts as both the primary sealing lip

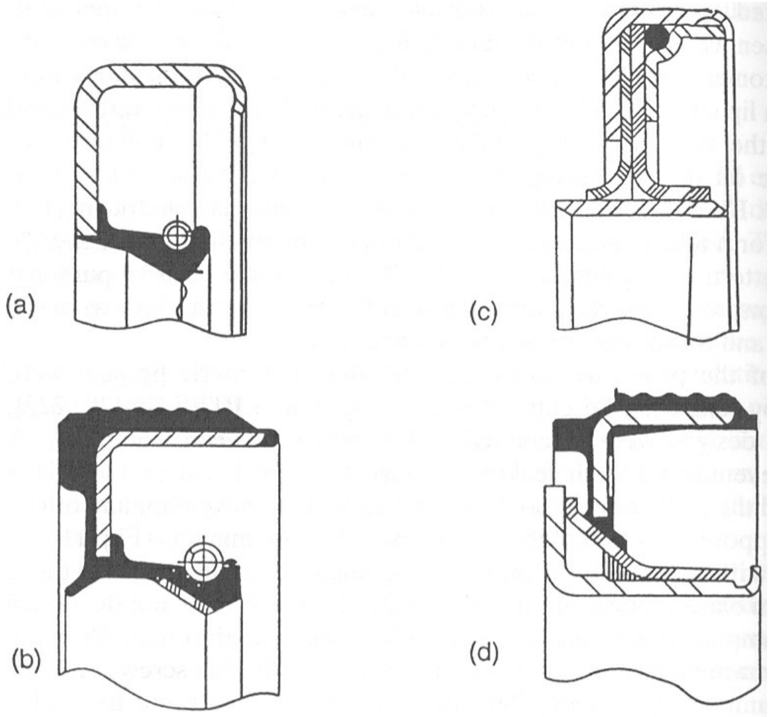


Fig. 11.2 (a) Conventional silicone or fluoroelastomer radial lip seal. (b) Primary lip lined with PTFE. (c) Assembled PTFE seal. (d) Bonded unitized PTFE seal. (Courtesy of Chicago Rawhide.)

and a secondary axial contaminant exclusion lip. The seal is assembled over a flanged wear sleeve that is curled over to permanently unitize the seal assembly and protect the sealing surfaces against damage from handling and installation. The resulting bonded package has fewer components than the assembled seal, no internal leakage path, a unitized wear sleeve, and an axial dirt lip rather than a radial one. Tests show that the axial dirt lip is superior to the radial design.

The PTFE wafer seal has a lower underlip temperature (Fig. 11.3) than either the PTFE laminate or the fluoroelastomer lip seal designs even though the torque and power consumption are greater (Figs. 11.4 and 11.5). This apparent paradox can easily be explained. The fluoroelastomer and PTFE laminated seals have a very narrow contact band (0.25 mm; 0.010 in.). The heat generated by the frictional torque is dissipated over this small area. The PTFE wafer seal has a higher frictional torque, but the heat is spread out

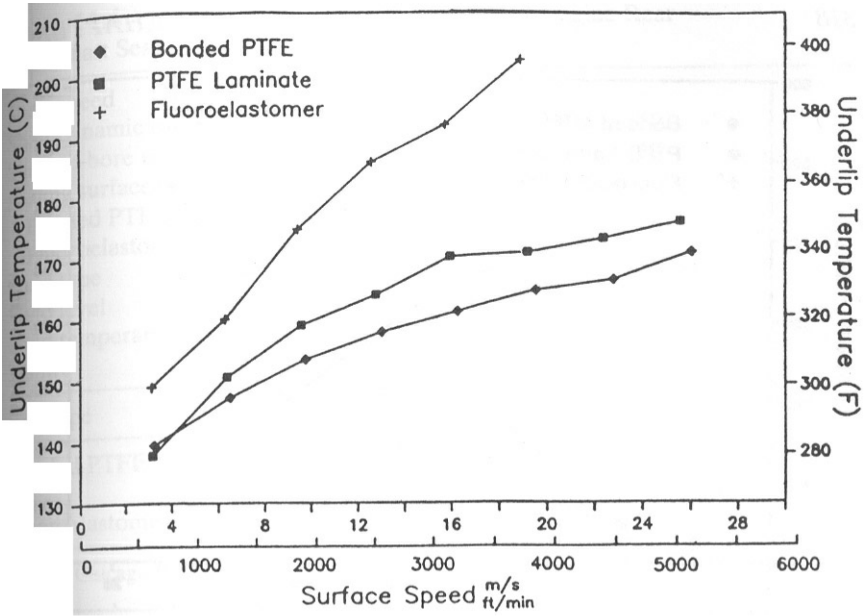


Fig. 11.3 Underlip temperature versus shaft surface speed. (Courtesy of Chicago Rawhide.)

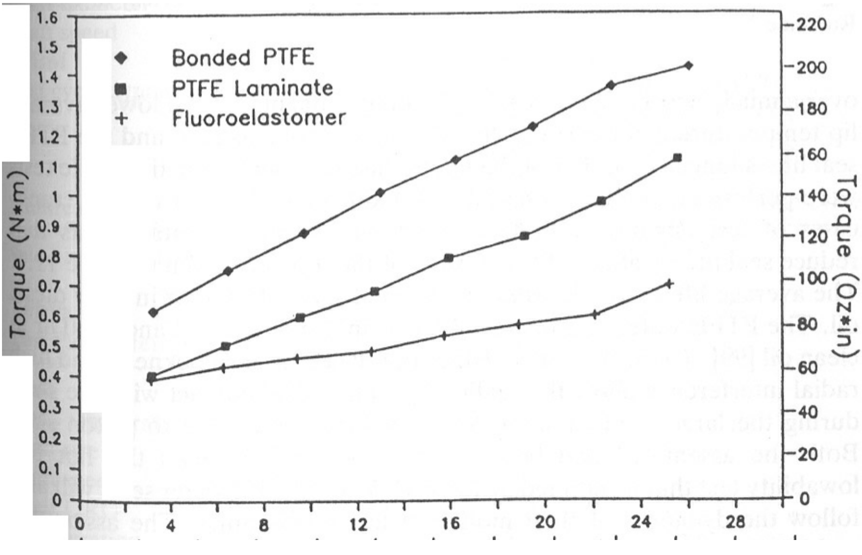


Fig. 11.4 Seal torque versus shaft surface speed. (Courtesy of Chicago Rawhide.)

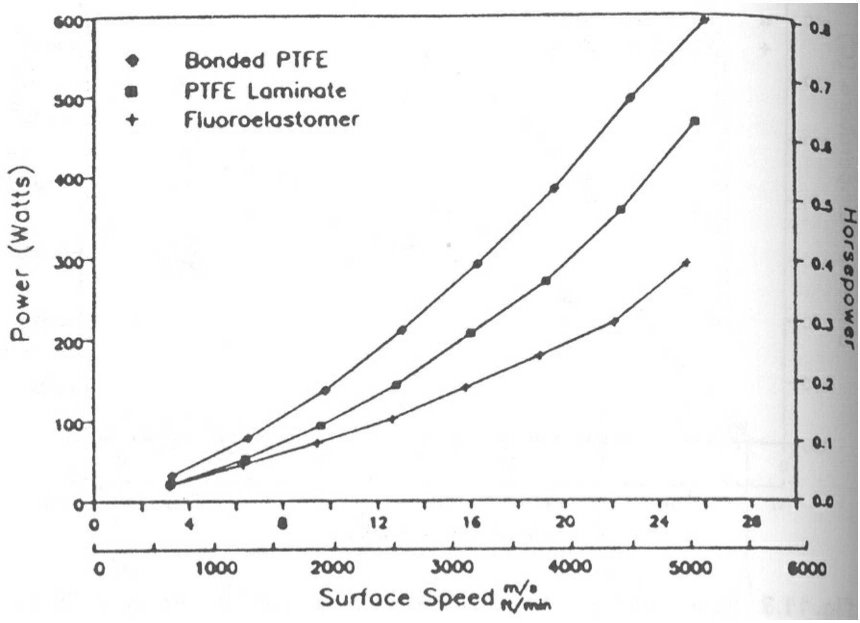


Fig. 11.5 Power consumption versus shaft surface speed. (Courtesy of Chicago Rawhide.)

over a much larger contact region (2.5 mm; 0.100 in.), and a lower average lip temperature is the result. The oil does not coke as fast, and the PTFE seal life is longer than that of the fluoroelastomer seal. Functional life tests were performed in dirty diesel oil that had been collected from the crankcases of test engines. The dirty diesel oil contained contaminants that reduce seal life by about 40%. Results of the tests are given in Table 11.4. The average life of the fluoroelastomer seal was only 188 hr in dirty diesel oil. The PTFE wafer seal life was 1268 hr in dirty diesel oil and 2060 hr in clean oil [99]. The wide contact band, thin PTFE wafer thickness, and high radial interference allow the sealing lip to maintain contact with the shaft during the large random shaft deflections that occur due to piston slap. Both the assembled and bonded PTFE wafer seals pass the hot followability test that is outlined in Table 11.5. The PTFE wafer seals will also follow the dynamics of shaft motion at low temperature. The assembled and bonded PTFE wafer seals passed the cold followability test described in Table 11.6.

Diesel engines also have a front crankshaft seal and camshaft seal that are usually smaller than the rear. An auxiliary drive seal application is also

Table 11.4 Functional Life Bench Tests--Diesel Engine Rear Crankshaft Seals

Shaft speed	37000 rpm
Shaft dynamic runout	0.010 in. (0.25 mm) TIR
Shaft-to-bore misalignment	0.020 in. (0.51 mm) TIR
Seeling surface diameter	
Bonded PTFE	4.900 in. (124.46 mm)
Fluoroelastomer	4.740 in. (120.4 mm)
Fluid type	Torqgard 30, clean and dirty
Fluid level	One-eighth shaft
Fluid temperature	280°F (138°C)
Failure	Measurable leakage

Seal type	Oil condition	Average life (hr)
Bonded PTFE	Clean	2060
	Dirty	1268
Fluoroelastomer	Dirty	188

Source: Chicago Rawhide.

Table 11.5 Hot Followability Test Parameters

Fluid	Delco SAE 10W
Sump temperature	121°C (250°F)
Shaft diameter	124.5 mm (4.900 in.)
Shaft speed	3700 rpm
STBM	0.51 mm (0.020 in.) TIR
Test cycle/dynamic runout	200 hr at 0.75 mm (0.030 in.) TIR 10 min at 1.00 mm (0.040 in.) TIR 10 min at 1.25 mm (0.050 in.) TIR 10 min at 1.50 mm (0.060 in.) TIR
Failure criteria	Any leakage during any part of the cycle

Source: Chicago Rawhide.

Table 11.6 Cold Followability Test Parameters

Fluid	Mobil 1 synthetic oil[a]
Sump temperature	-46°C (-50°F)
Shaft size	124.5 mm (4.900 in.)
Shaft speed	60 rpm
Shaft DRO	0.25 mm (0.010 in.) TIR
Test duration	30 min
Failure criteria	Leakage or lip fracture

[a] Low-viscosity at low temperatures.

Source: Chicago Rawhide.

often found. These applications are better lubricated, with lower shaft surface speed and lower lip temperatures, than the rear crankshaft seal. As a result, they do not exhibit as many leakage problems as the rear. PTFE is typically used for seals in these areas.

11.3 AUTOMOTIVE ENGINE CRANKSHAFT SEALS

Automotive engines have rear and front crankshaft and camshaft seals like a diesel engine. The evolution of the automotive engine has proceeded on a slightly different path that that of the diesel engine, and the sealing solution is different. Early engines were large and operated at slow speeds (a maximum of 3600 rpm). Oil temperature was relatively low, and the oils were mild with few harsh additives. The consumer did not register complaints if some leakage occurred. The early rear crankshaft seals were made of a split impregnated cork material. The front and cam seals were typically made of a nitrile. Engine speeds increased, and the rear seals were replaced with a silicone material. Early seal designs consisted of two unsprung silicone seal halves that were matched around the shaft (Fig. 11.6). This design was later replaced by a full spring-loaded seal with no splits. Silicone was the material of choice for the rear crankshaft application, and a nitrile or polyacrylate was used for the front crankshaft and camshaft.

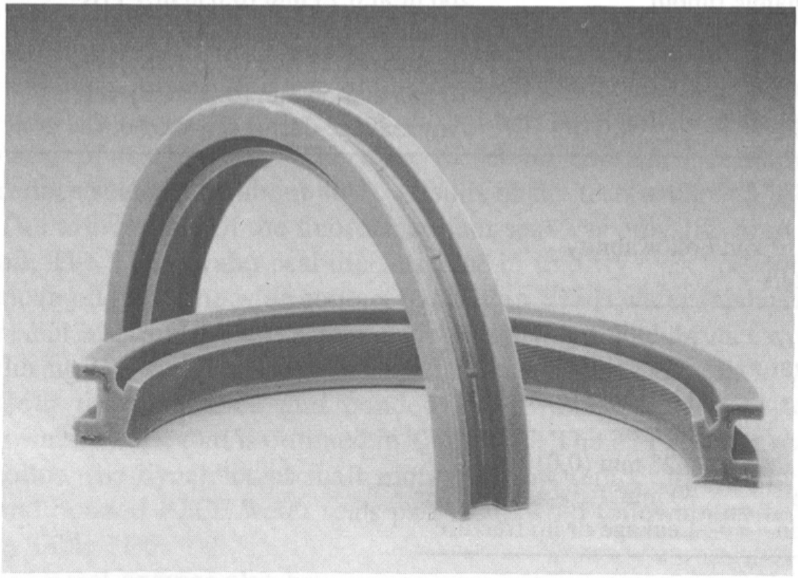


Fig. 11.6 Automotive half seals for rear crankshaft applications.

In recent years, automobile manufacturers have been designing smaller engines with greater horsepower for fuel economy purposes [12, 60, 63]. The increased speed of the smaller engines, automobile downsizing, and reduced engine compartment space have raised underhood temperatures considerably. Additive packages in the latest classification of automobile engine oils have been designed to reduce engine wear and increase bearing life. The higher temperatures and oil additive changes have an increasingly destructive effect on silicone, causing the material to revert to its gummy state at a faster rate, leading to shorter seal life. This problem was solved by replacing the silicone material with fluoroelastomer [126]. The fluoroelastomer resists the chemicals contained in the modern oil additive package. Additional studies revealed that seal failure with fluoroelastomer was often caused by carbonized oil coating the wear track region, covering the microasperities and destroying the natural pumping ability of the seal. Seal design was changed to reduce load and underlip temperature and thus minimize the coking problem [45]. Lower modulus fluoroelastomer compounds also reduced seal load and underlip temperature [100].

The rear crankshaft seals are often pressed into an aluminum retainer, which is bolted to the engine block (Fig. 11.7). The seal design has evolved from a metal O.D. to a full rubber O.D. (Fig. 11.8a) and now a half rubber, half metal O.D. is often recommended (Fig. 11.8b). The half rubber, half metal O.D. has advantages over the full rubber O.D. that are listed in Table 11.7 [46, 100]. If the bore does not contain a generous chamfer or radius to facilitate installation of the seal or if it contains a sharp edge, the full rubber O.D. can be sheared off during insertion of the seal. The half rubber, half metal design resists this problem because the outer edge has a metal chamfer that serves as a pilot during installation. When a full rubber O.D. is installed into the bore, the axial frictional force at the rubber O.D. causes the rubber wall to deform due to shear stress. Upon removal of the installation load, the rubber O.D. wall returns to an equilibrium position, resulting in springback out of the bore (Fig. 11.9). Nonuniform springback along the periphery of the O.D. results in misalignment or cocking of the seal relative to the bore face; this can cause distortion of the seal lip and result in premature leakage. The metal portion of the half metal, half rubber design acts as a positive lock during installation and minimizes springback. When full rubber O.D. seals are molded, rubber flows up the inside wall of the seal case to fill the rubber O.D. cavity. During this process, high pressures are generated within the mold cavity at the inner wall of the seal case, which can distort the case outward as shown in Fig. 11.10, resulting in a nonuniform rubber O.D. and possible unfills at the O.D. The half rubber, half metal O.D. eliminates this problem because the metal case is supported by direct contact with the mold wall. Another

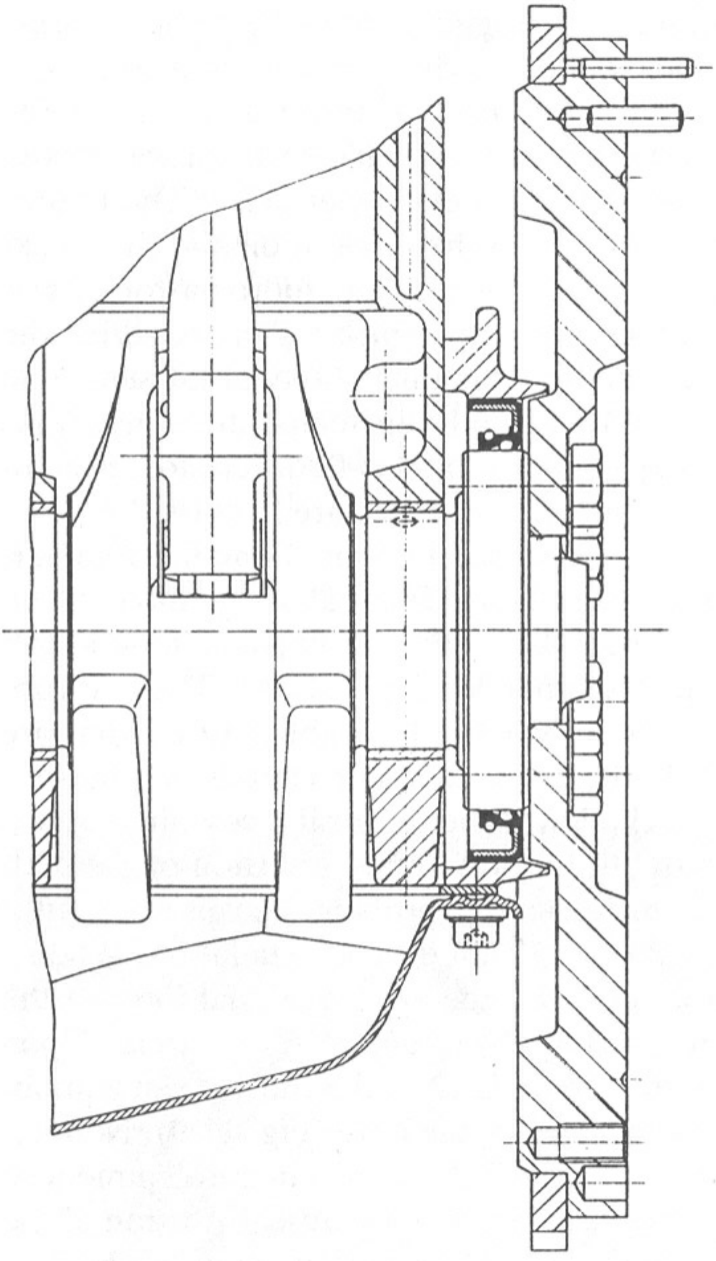


Fig. 11.7 Automotive engine rear crankshaft sealing application. Shaft diameter, 76.2--101.6 mm (3.000--4.000 in.); maximum shaft speed, 6000 rpm; lubricant, SG, SH engine oils, maximum sump temperature 150°C (302°F).

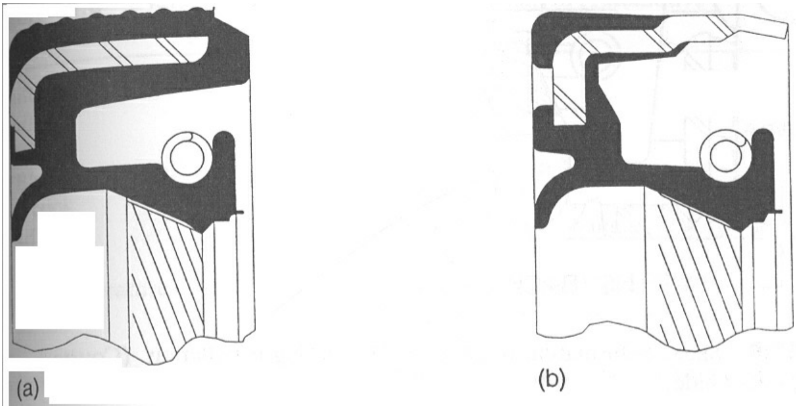


Fig. 11.8 Automotive engine crankshaft seals. (a) Full rubber O.D. (b) Half rubber, half metal O.D. (Courtesy of Chicago Rawhide.)

Table 11.7 Advantages of Half Rubber, Half Metal O.D. over Full Rubber O.D.

1.	Provides installation pilot into the bore.
2.	Eliminates shearing damage of rubber O.D.
3.	Reduces installation springback (full rubber, 0.055 in. vs. half rubber, 0.008 in.).
4.	Reduces seal cocking in bore (full rubber, 0.008 in. vs. half rubber, 0.004 in.).
5.	Reduces installation force (full rubber, 1500 lb vs. half rubber, 980 lb).
6.	Combination metal and rubber press fit maintains high retention forces in aluminum bores after 1500 hr oil heat aging at 300°F (760 lb).
7.	Eliminates metal case distortion due to molding pressures.
8.	Minimizes rubber waste during molding.

Source: Chicago Rawhide.

benefit is realized because the rubber flash at the O.D. of the all rubber O.D. is eliminated, which results in a material savings.

Seal life tests were run to demonstrate the performance improvements realized over a 10 year development period [100]. The results are summarized in Table 11.8. The reduced modulus fluoroelastomer with the half rubber, half metal O.D. and reduced lip load had the longest life. It is anticipated that the next generation of automotive engine seals will be value added [30]. The seal, retainer, and gasket will be combined into a

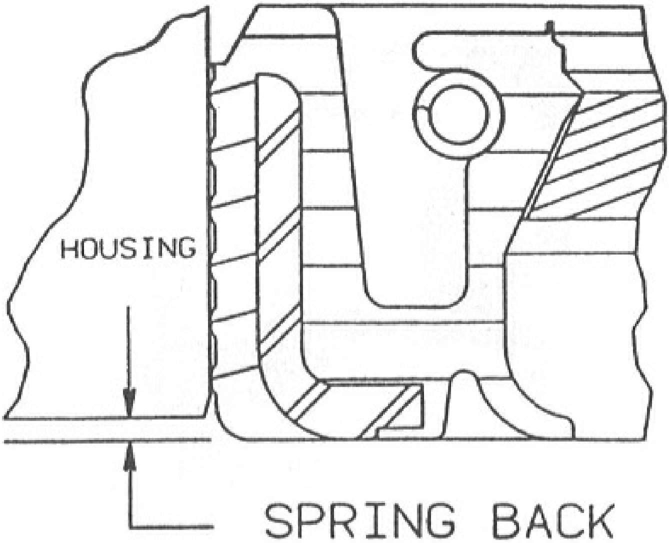


Fig. 11.9 Shear deformation of rubber O.D. during installation. (Courtesy of Chicago Rawhide.)

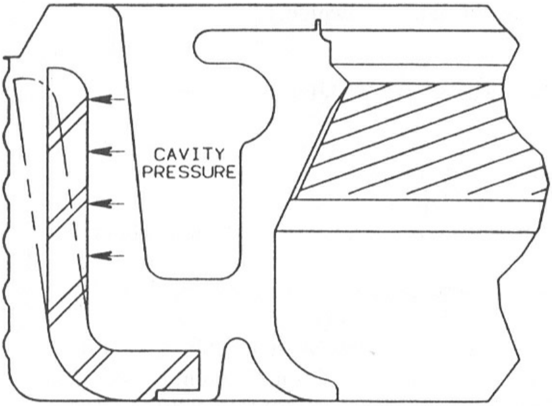


Fig. 11.10 Molding pressure can distort the seal case. (Courtesy of Chicago Rawhide.)

single package for the rear crankshaft application (Fig. 11.11). The end user will thus have one component instead of three to assemble. Lightweight thermoplastic materials are being considered as a replacement for the aluminum retainer [124]. Research and development are under way to reduce seal torque and power consumption [61, 100, 136].

The automotive front crankshaft, camshaft, and auxiliary shaft seal applications are milder than those of the rear crankshaft and have smaller shafts, lower lip temperatures, and longer life (Fig. 11.12). Fluoroelastomer is the recommended material.

Table 11.8 Evolution of Performance Improvement in Automotive Rear Crankshaft Applications

Shaft speed	6000 rpm
Shaft size	92 mm (3.625 in.)
DRO	0.25 mm (0.010 in.) TIR
STRM	0.50 mm (0.020 in.) TIR
Seal cock	0.25 mm (0.010 in.)
Fluid type	SAE 10W30 (SF), change every 100 hr
Fluid level	Centerline
Sump temperature	8 hr at ambient and 0 rpm, 8 hr at 149°C (300°F) and 6000 rpm, 8 hr at 132°C (270°F) and 6000 rpm
Failure criteria	1 or more leakage in one day or a total of 3 gs accumulation

Material	Seal design	Seal life (hr)
Silicone	Full rubber O.D., conventional lip	622
Fluoroelastomer	Full rubber O.D., conventional lip	1996
Reduced modulus fluoroelastomer	Full rubber O.D., conventional lip	3162
	Half rubber O.D., reduced lip load	4004

Source: Chicago Rawhide.

11.4 PINION SEALS

Many automobiles and light and heavy trucks still have rear wheel drive. A driveshaft connects the transmission to the differential. The differential has gears that change the high-speed rotation from the transmission to a low-speed rotation that is imparted to the drive axles that turn the wheels to propel the car. The differential will have a seal at the front where the high-speed rotation from the driveshaft yoke enters the housing (Fig. 11.13). This seal is commonly called a pinion seal because it is located close to the pinion gear in the differential. It is sometimes referred to as an input seal. It must withstand the aggressive additive packages that are placed in the gear oil to protect the gears. The seal is exposed to both low and high temperatures and must operate over a range of shaft speeds that can become relatively high. It is located in a position that is constantly bombarded with water, dust, and mud. Axial end play can also be expected in the application. Vehicle manufacturers are continuing to extend warranties, which means that modern seals must last longer than older designs. Commercial vehicles currently have warranty periods of 804,500 km (500,000 mi), and manufacturers are demanding increases to 1,609,000 km

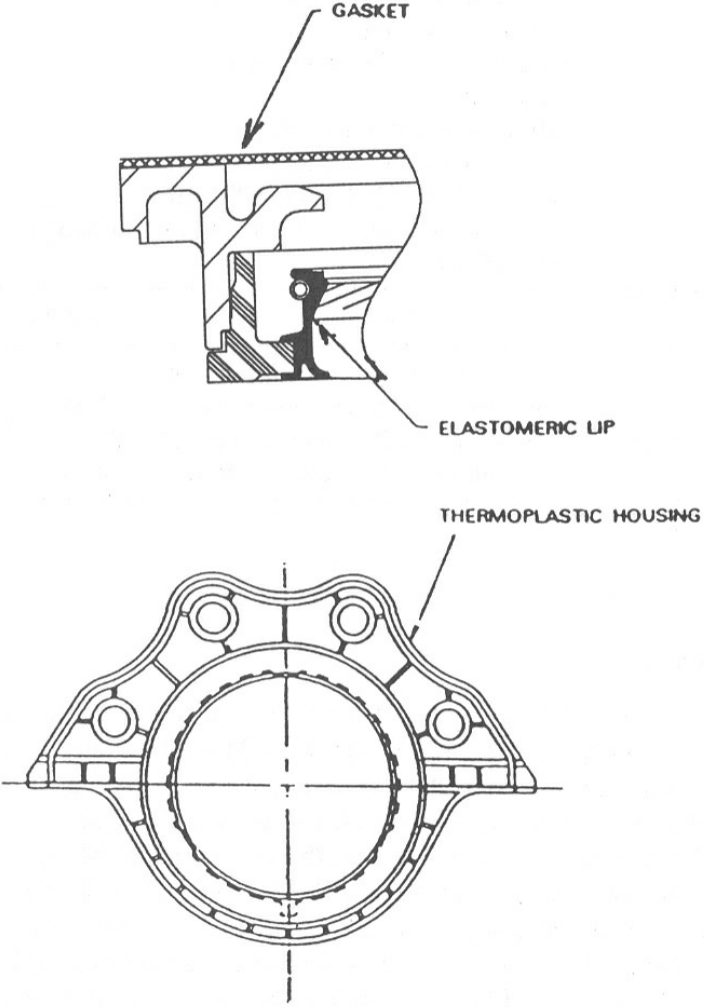

GASKET
ELASTOMERIC LIP
THERMOPLASTIC HOUSING

Fig. 11.11 Automotive rear crankshaft seal composite application. (Courtesy of Chicago Rawhide.)

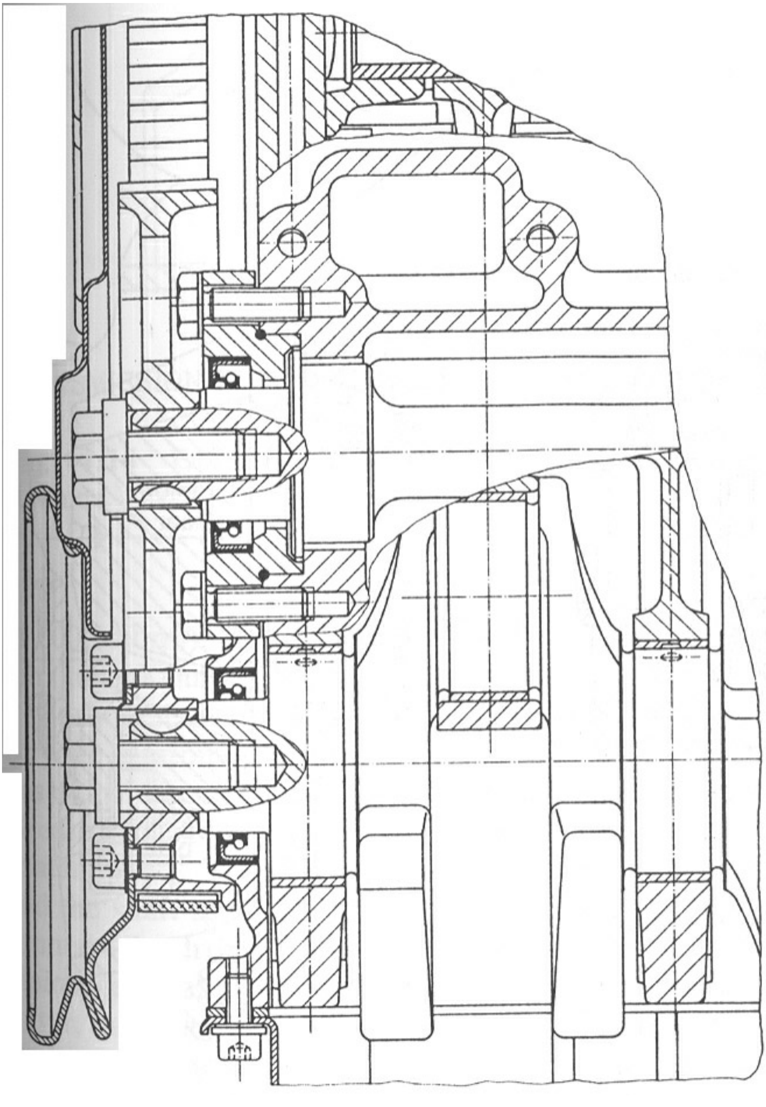


Fig. 11.12 Automotive front crankshaft seal application. Shaft diameter, 30--40 mm (0.181--1.574 in.); maximum shaft speed, 6000 rpm; lubricant, engine oil; maximum temperature, 130°C (266°F). (Courtesy of Chicago Rawhide.)

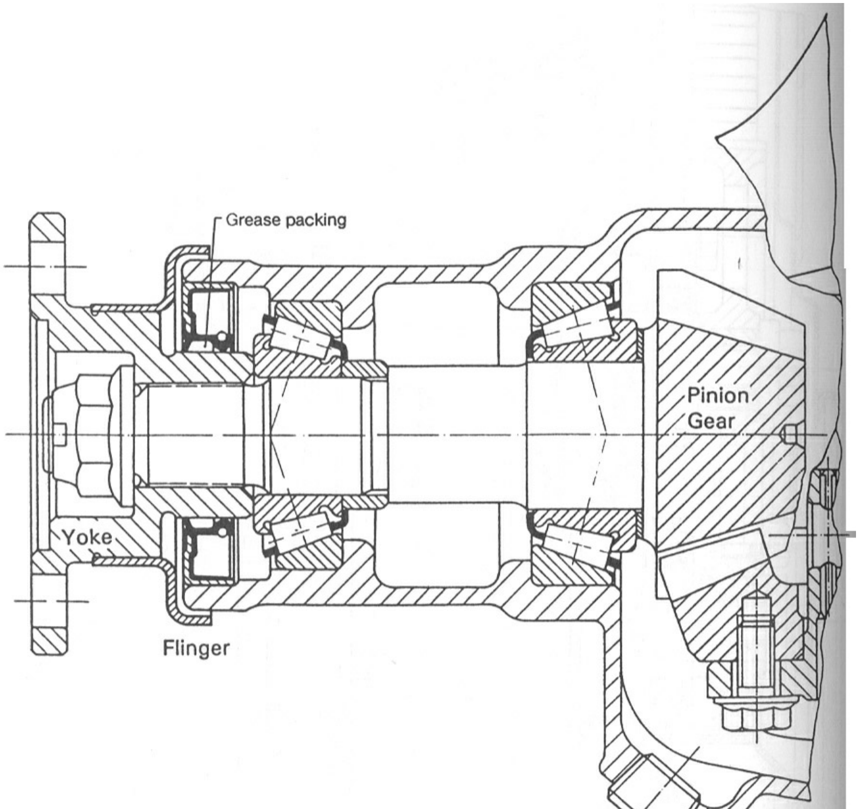

Grease packing
Pinion
Gear
Yoke
Flinger

Fig. 11.13 Pinion seal application. Approximate seal size, 50 mm (1.968 in.); maximum shaft speed, 6000 rpm; lubricant, hypoid or other gear oil; maximum temperature, 160°C (230°F); external environment--dust, water, mud. (Courtesy of Chicago Rawhide.)

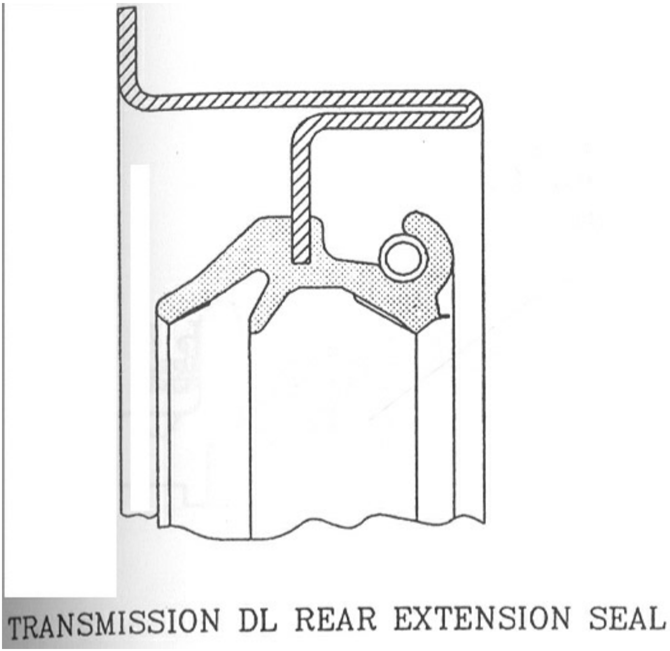


Fig. 11.14 Extended dirt lips for pinion seal application. (Courtesy of Chicago Rawhide.)

(1,000,000 mi). Automotive and light truck manufacturers want 160,900 km (100,000 mi).

Early seals were of the traditional dual-lip design with nitrile or polyacrylate elastomeric compounds. These were replaced with a design that has a dual extended lip (Fig. 11.14), and the elastomer is fluoroelastomer. Grease is packed between the extended dirt lip to serve as a barrier to the contaminants. The seal is typically installed into the differential housing, and the shaft sealing surface will engage the seal lips when the yoke that connects the drive shaft to the pinion gear is slipped over the splines on the pinion gear shaft. A flinger is sometimes installed on the yoke to deflect water, mud, and dust away from the sealing area. If contaminants do pass the barriers, then heavy yoke and seal lip wear can result, which will lead to early leakage. Often both the seal and the yoke must be replaced, which is a costly project. A unitized seal design (Fig. 11.15) maximizes contaminant exclusion while protecting the yoke surface from damage [13].

Two axial dirt lips are incorporated in the design to provide the most effective method of excluding contaminants. A PTFE thrust bumper is used to minimize the friction between the flange and the seal after installation. The bumper also acts as a labyrinth, and the flange acts as a flinger to reduce contaminant ingestion. The wear sleeve contains the running surface for the seal; thus the yoke can be reused if the seal must be replaced.

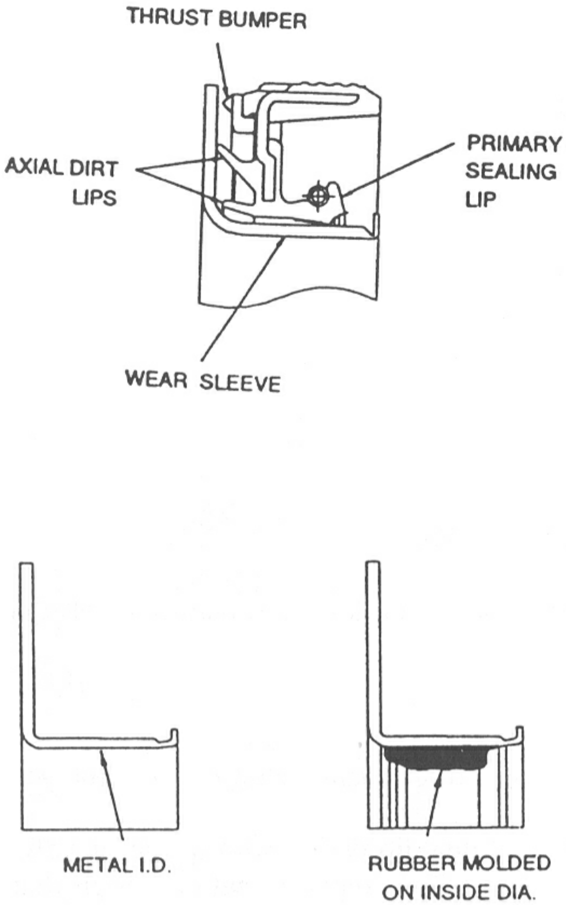


Fig. 11.15 Unitized pinion seal. (Courtesy of Chicago Rawhide.)

The wear sleeve is curled to protect the seal lip and the wear sleeve running surface and prevent disassembly of the unitized seal package. If the seal is installed in the differential housing bore, then the sleeve should have rubber molded on the inside diameter. This is not recommended for high-speed applications because the rubber impedes heat transfer. The underlip temperature will be higher than for a seal with a full metal sleeve, and the seal's ultimate life may be shortened. The installation of the metal I.D. seal is different than for the rubber I.D. seal. The sleeve is first installed on the yoke with an assembly tool that positions the seal correctly. The yoke and seal are then fitted over the pinion shaft (Fig. 11.16). The PTFE thrust bumper absorbs the thrust load that occurs during assembly. Typical bench test conditions for life tests and environmental tests for pinion application appear in Tables 11.9 and 11.10.

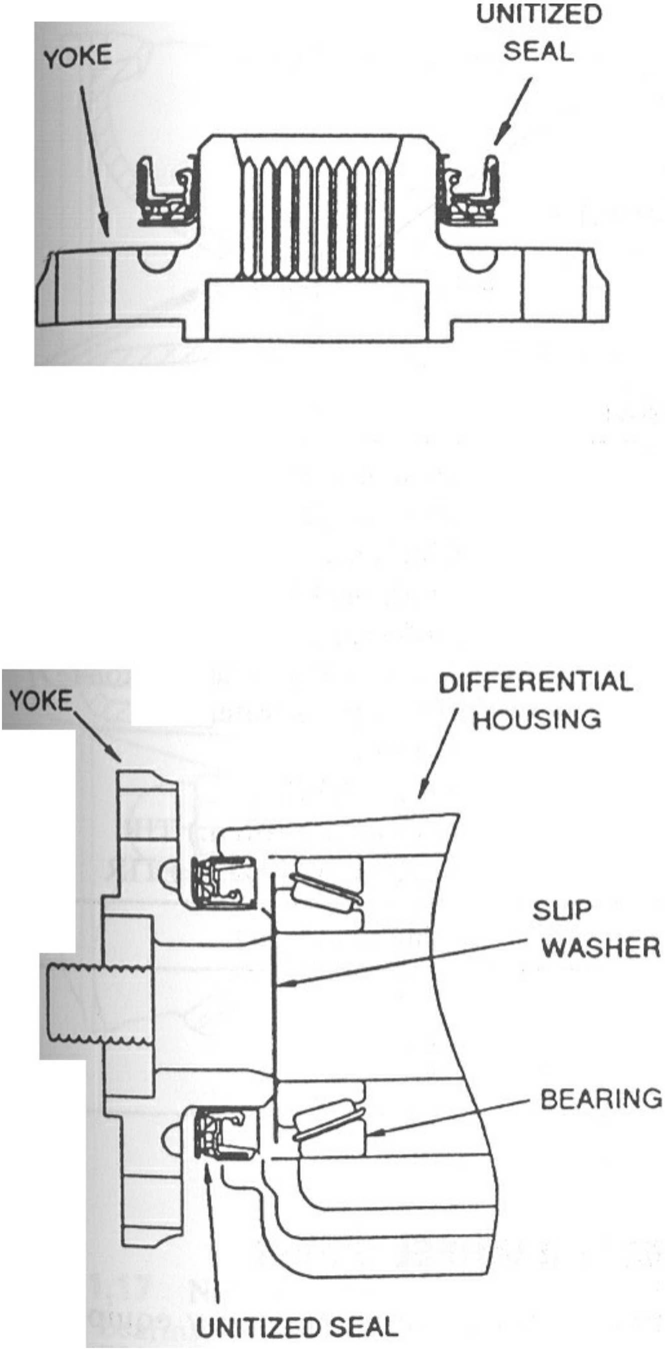

YOKE
UNITIZED SEAL
YOKE
DIFFERENTIAL HOUSING
SLIP WASHER
BEARING
UNITIZED SEAL

Fig. 11.16 Installation of metal I.D. pinion seal. (Courtesy of Chicago Rawhide.)

Table 11.9 Typical Pinion Seal Life Test Conditions for Bench Testing

Shaft size	40--50 mm (1.574--1.968 in.)
Bore size	50--60 mm (1.968--2.362 in.)
Rotation	Clockwise
Cycle	7 hr on, 1 hr off, 5 min reversal every 500 hr
Lubricant	Differential gear
Shaft speed	2900 rpm
Sump temperature	121°C (225°F)
DRO	0.25 mm (0.010 in.)
STBM	0.127 mm (0.005 in.)

Source: Chicago Rawhide.

Table 11.10 Mud Slurry Test Results

Shaft size	46 mm (1.812 in.)
Bore size	70.5 mm (2.779 in.)
Rotation	Clockwise
Cycle	20 hr on, 4 hr off
Lubricant	Differential gear
Slurry mixture	3 parts sand, 1 part bentonite clay, 8 parts water
Shaft speed	100 rpm
Temperature	79°C (175°F)
DRO	0.25 mm (0.010 in.) TIR
STBM	0.127 mm (0.005 in.) TIR
Seal type	**Time to fail (hr)**
Conventional radial exclusion lip	829
Dual extended exclusion radial lip	948
Unitized axial exclusion lip	1293

Source: Chicago Rawhide.

11.5 TRUCK AND AUTOMOTIVE WHEEL SEALS

Wheel seals are used on automotive, truck, and off-highway equipment axles to retain grease or oil and exclude contaminants [13, 35, 107, 138, 230, 231]. Many light truck and automotive applications use greased bearings for the nondriven axles. Conventional radial lip seals can be used in these applications. Some seals are spring-loaded (Fig. 11.17), but other seals are not. The elastomeric beam section is deflected and provides the required loading on the shaft (Figs. 11.18a, 11.18b). The best results for a

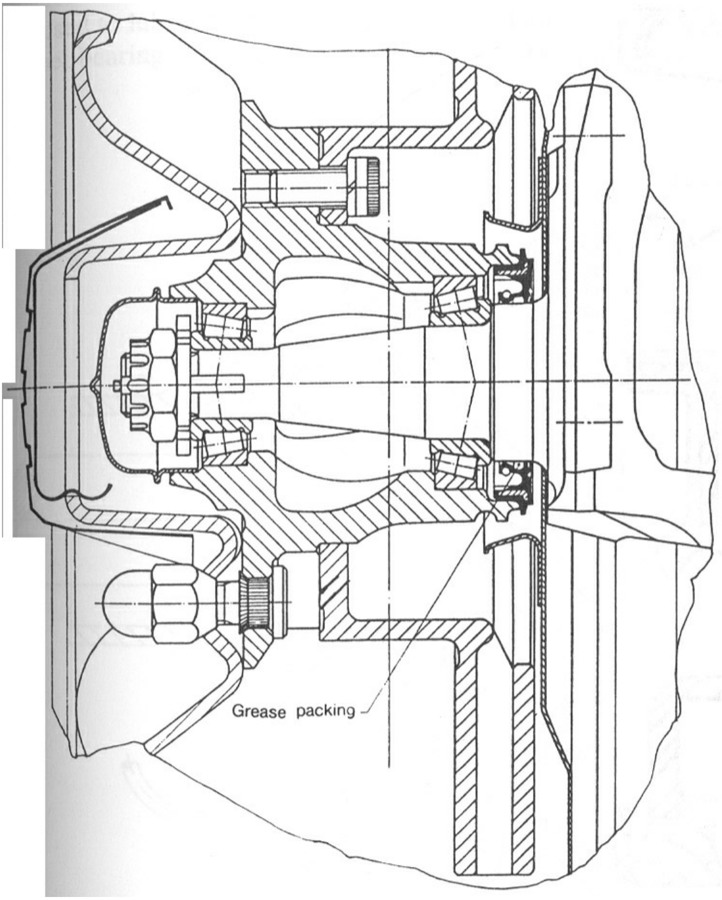


Fig. 11.17 Nondriven automotive and truck wheel seal application--grease-packed bearings. Shaft diameter, 38--152 mm (1.500--6000 in.); maximum shaft speed, 750 rpm; lubricant, grease; maximum temperature, 80°C (175°F). (Courtesy of Chicago Rawhide.)

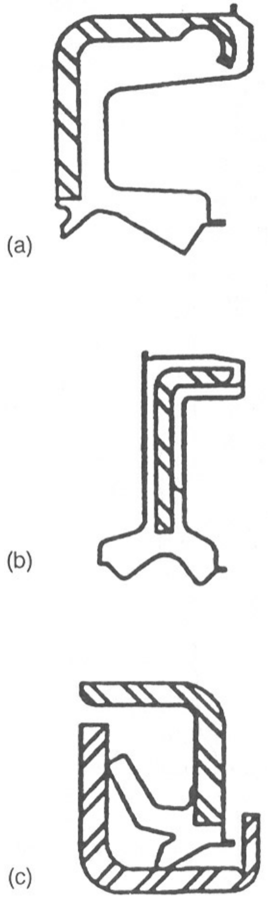


Fig. 11.18 Nonspring-loaded seal designs for greased truck wheel seal bearings. (a) Dual radial lips, rubber O.D. on inside wall, and nose gasket. (b) Rubber-encased seal to prevent corrosion, (c) Unitized seal; axial lip faces contaminants; stainless steel wear sleeve. (Courtesy of Chicago Rawhide.)

greased bearing application occur with a seal design that has an axial dirt lip that rides on a vertical surface to prevent entry of contaminants that will shorten seal life (Fig. 11.18c).

Automotive or light truck drive axles usually have conventional spring-loaded seals. In some cases, the sealing configuration prevents the hypoid oil used to lubricate the gears in the differential from entering the greased wheel bearings (Fig. 11.19). Many heavy truck applications lubricate the

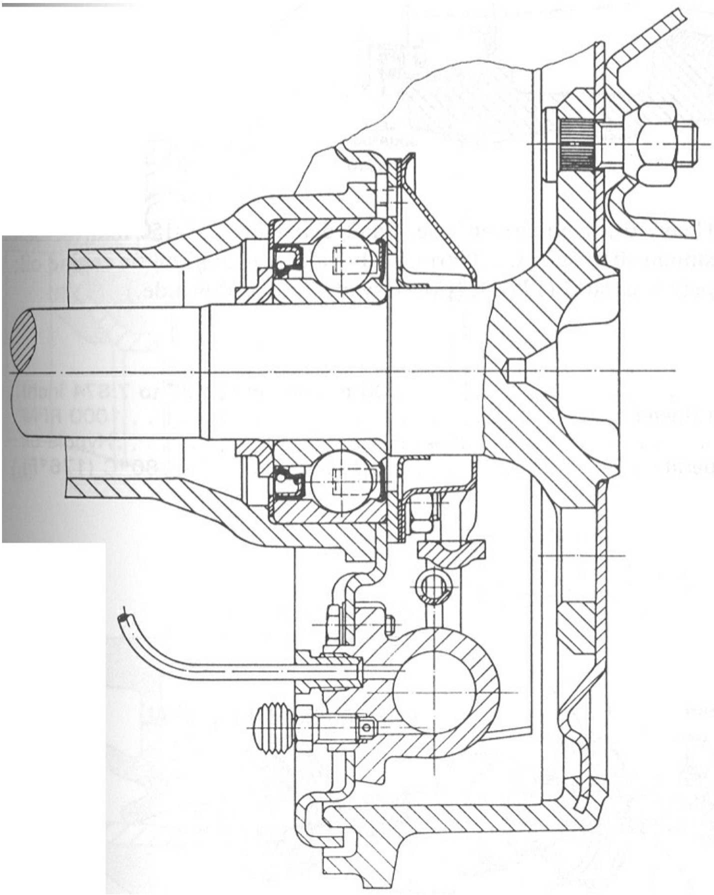


Fig. 11.19 Driven axle wheel seal grease bearing for automobiles and light trucks. Shaft diameter, 40--100.0 mm (1.574--3.937 in.); maximum shaft speed, 1500 rpm; lubrication, hypoid oil or grease; maximum temperature, 80°C (176°F). (Courtesy of Chicago Rawhide.)

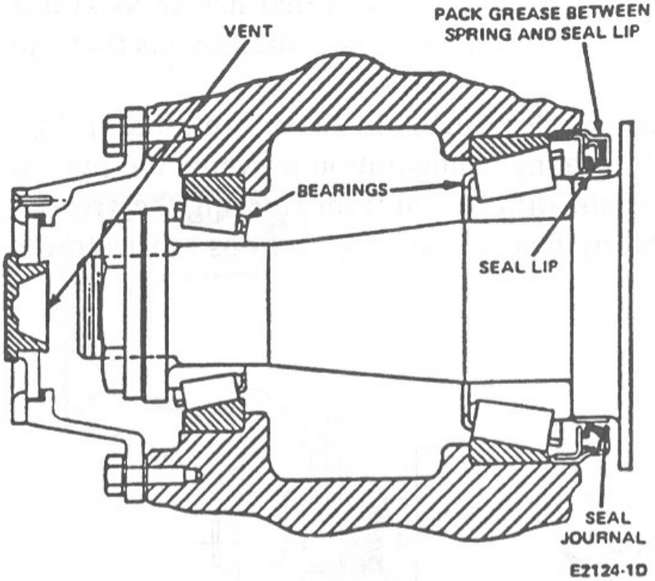

VENT
PACK GREASE BETWEEN SPRING AND SEAL LIP
BEARINGS
SEAL LIP
SEAL JOURNAL
E2124-1D

Fig. 11.20 Heavy truck nondriven axle. Shaft diameter, 100--150 mm (3.936--5.906 in.); maximum shaft speed, 1000 rpm; lubrication, hypoid oil or engine oil; maximum temperature, 80°C (175°F). (Courtesy of Chicago Rawhide.)

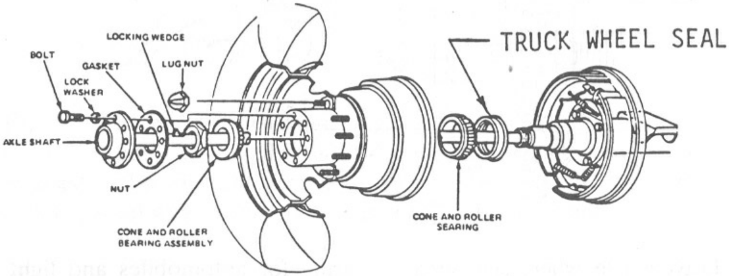

LOCKING WEDGE
BOLT
GASKET
LUG NUT
LOCK WASHER
AXLE SHAFT
NUT
CONE AND ROLLER BEARING ASSEMBLY
TRUCK WHEEL SEAL
CONE AND ROLLER BEARING

Fig. 11.21 Heavy truck driven axle. Shaft diameter, 100--200 mm (3.937--7.874 in.); maximum shaft speed, 1000 rpm; lubrication, hypoid oil; maximum temperature, 80°C (176°F). (Courtesy of Chicago Rawhide.)

bearings of both nondriven (Fig. 11.20) and driven (Fig. 11.21) axles with oil. The seals used in these applications are usually unitized. The hub is rotating and the spindle is stationary in these applications. The external seal design (Fig. 11.22a) has a stationary seal element with a rotating sealing surface. The internal design (Fig. 11.22b) has a rotating seal with a stationary sleeve. This seal has rubber on both the inside and outside to allow hand installation in the field without special tools and has an axial dirt lip with a flinger for maximum contaminant exclusion [138].

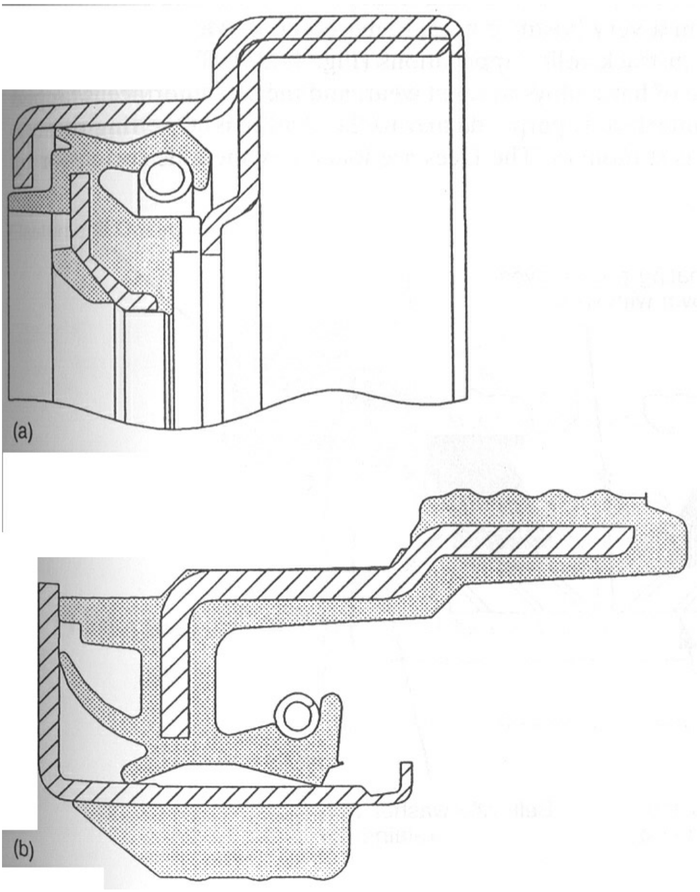


Fig. 11.22 Unitized truck wheel seals. (a) Stationary packing, rotating sealing surface, external seal; requires installation tools. (b) Rotating packing, stationary sealing surface; hand installations; axial dirt lip and flinger. (Courtesy of Chicago Rawhide.)

Automotive and light truck wheel seals are usually made with a nitrile or Vamac material. Truck wheel seals are also made from nitrile, but more aggressive lubricants, increasing use of synthetic lubes, and higher application temperatures are resulting in a slow switch to more expensive materials [35].

11.6 TRACKED VEHICLES

Tracks for construction equipment and tanks are driven by rollers, and idler rollers guide the chain and hold it in place. These rollers are lubricated, and some means of preventing leakage and excluding contaminants must be used in a very hostile external environment. Metal face seals are typically used in track roller applications (Fig. 11.23). The metal components are made of hard alloys to resist wear, and the seal interface is lapped flat. The seal interface is perpendicular to the shaft axis. One ring rotates, and the other is stationary. The faces are loaded by metallic springs or by

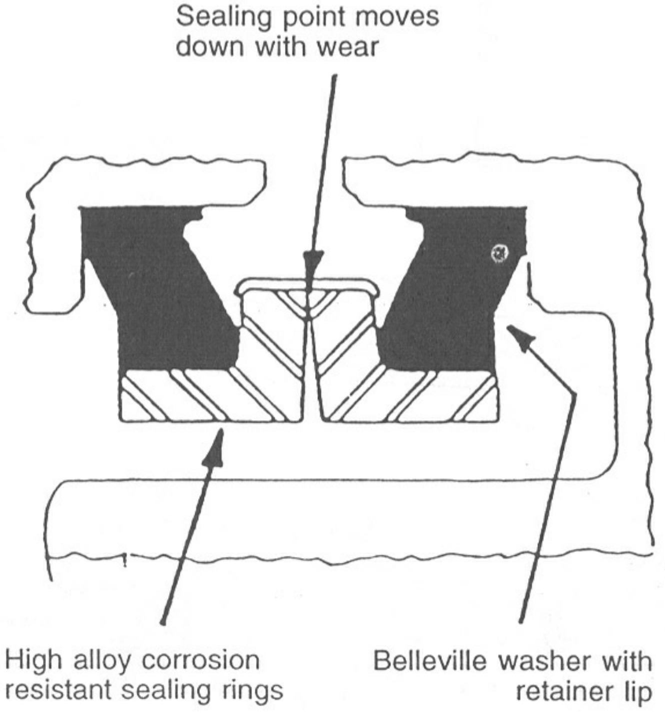


Fig. 11.23 Metal face seal for track roller. Variable shaft size. Maximum shaft speed, 3.8 m/sec (12.5 ft/sec); maximum internal pressure, 345 kPa (50 psi); lubricant, SAE 10W40 (centerline fill); temperature range, --54 to 107°C (--65 to 225°F). (Courtesy of Chicago Rawhide.)

rubber components that can act as springs. The rubber material is usually nitrile. Metal face seals can also be used on slow-moving vehicles with large wheels that are subjected to a harsh external environment. The metallic face seals replace the conventional lip seals that are normally used. It has also been proposed that metal face seals be used in the wet disk brake application for off-highway vehicles. The advantages of the wet disk brake, which is a totally sealed brake unit, when operating in harsh environments are low maintenance costs and longer brake life.

The links of the chain used to drive tracked vehicles are connected by pins that are often sealed with O-rings. Some track pins are lubricated with oil, and a more sophisticated design must be used to retain the oil and exclude contaminants. Figure 11.24 illustrates a face seal that is used on track pin applications. A polyurethane element is pressed against a vertical metal wall by a nitrile elastomeric component that acts as a spring. During operation, the seal oscillates back and forth fairly slowly.

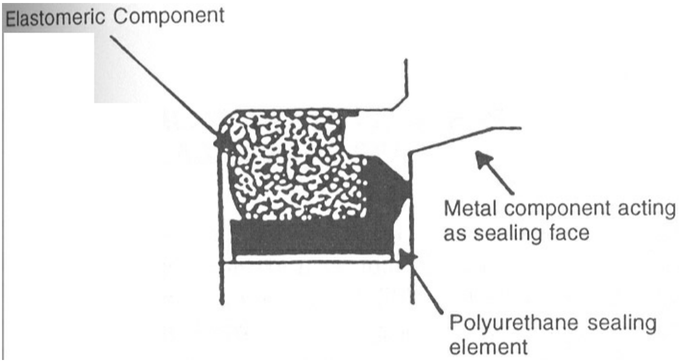

Elastomeric Component
Metal component acting as sealing face
Polyurethane sealing element

Fig. 11.24 Track pin seal. (Courtesy of Chicago Rawhide.)

12

Manufacturing Methods for Radial Lip Seals

12.1 OVERALL PROCESS FLOW FOR NONELASTOMERIC SEALS

Some radial lip seals are made from nonelastomeric materials such as PTFE or leather (208,209). A manufacturing process flow for a typical PTFE seal appears in Fig. 12.1. PTFE powder is mixed with other ingredients to create the desired seal compound. The powder is placed into a mold that is normally shaped like a sleeve with an I.D. slightly smaller and an O.D. slightly larger than the desired product. The powder is placed in the circular annulus that is formed by the mold, and the material is compacted under high pressure to a height that is about 20--25% that of the original fill. The green material is placed in an oven and sintered at a high temperature (375°C; 707°F). The resulting sleeve is shaped like a short, hollow cylinder or pipe. It is placed in a lathe and turned to the desired I.D. and O.D. A hydrodynamic spiral feature is sometimes cut into the face before the thin wafer is sliced off the billet.

The PTFE wafers are preformed over mandrels to form the sealing lips. The hydrodynamic element is sometimes stamped into the wafer during the performing operation. After the wafers have been performed, they are placed into the outer cup and a gasket or O-ring is added to prevent internal leakage. An inner cup is placed on top of the package, and the

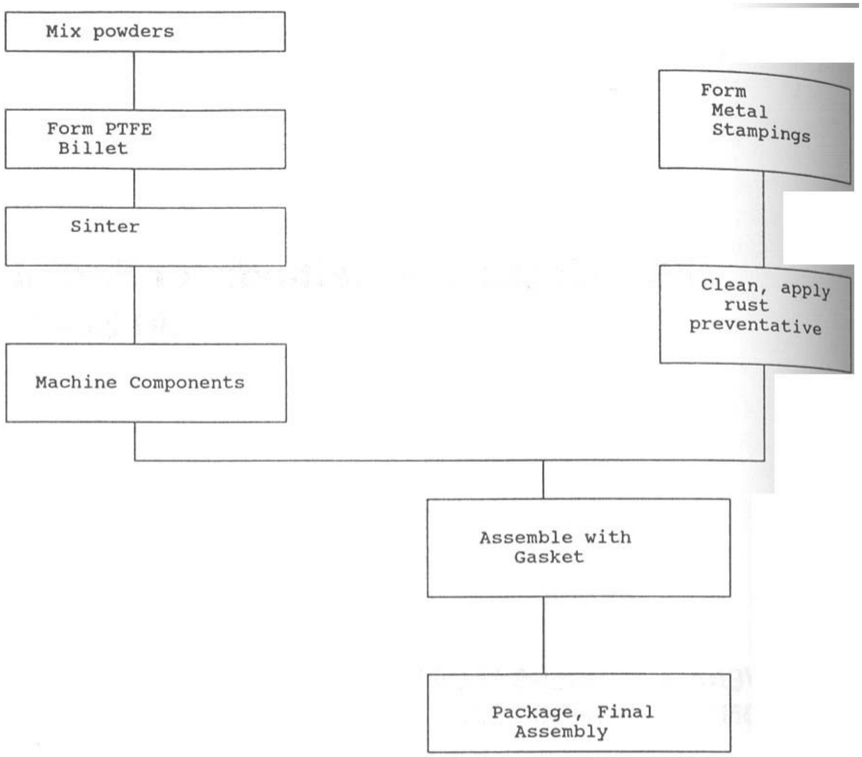

Mix powders
Form PTFE Billet
Sinter
Machine Components
Form Metal Stampings
Clean, apply rust preventative
Assemble with Gasket
Package, Final Assembly

Fig. 12.1 Process flow to manufacture nonelastomeric seals.

outer cup is rolled over to clinch the components together. Sometimes the seal is installed on a thin plastic sleeve to protect the PTFE lips during shipping and handling; this sleeve is removed during installation. The seal package is often combined with a wear sleeve, and the wear sleeve and the seal are shipped and installed together. A typical finished product appears in Fig. 12.2.

12.2 OVERALL PROCESS FLOW FOR ELASTOMERIC SEALS

The radial lip seal is a precision product that is manufactured to a high level of quality standards. The typical manufacturing method for an elastomeric seal is a batch process that begins with the weighing and mixing of raw ingredients to form the uncured compound (Fig. 12.3). The mixed material

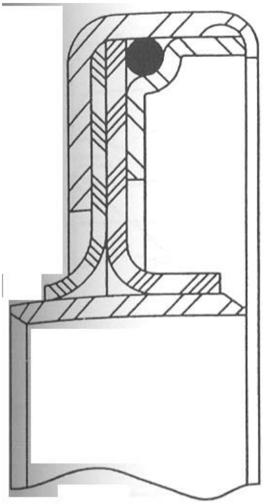


Fig. 12.2 Typical dual-lip PTFE seal on wear sleeve. (Courtesy of Chicago Rawhide.)

is quarantined until quality checks have been performed and the material is approved. If necessary, the material is cut into plugs or preps for the molding process. Metal stampings are made and cemented, and the preps are molded and bonded to the metal case during the vulcanization process. After molding, excess rubber flash is removed, trimming operations are performed, and the components are installed. When the seals are completed, they are inspected, packaged, and shipped to the customer. Details of the process used to manufacture elastomeric seals are discussed in the remainder of this chapter.

12.3 MIXING ELASTOMERIC COMPOUNDS

The rubber compounder prepares an oil seal recipe that may contain as many as 15 ingredients to provide the required application properties while maintaining good manufacturing capabilities. The materials must be mixed completely and be dispersed uniformly throughout the mass. The polymer is very stiff, and large quantities of energy must be added to the mix to masticate the materials, warm them up, and disperse the other ingredients throughout the mass. Several mixing methods can be used, but the internal mixer is preferred [16]. Two spiral rotors rotate within a housing, and the clearances between the blades of the rotor and the housing are adjusted to

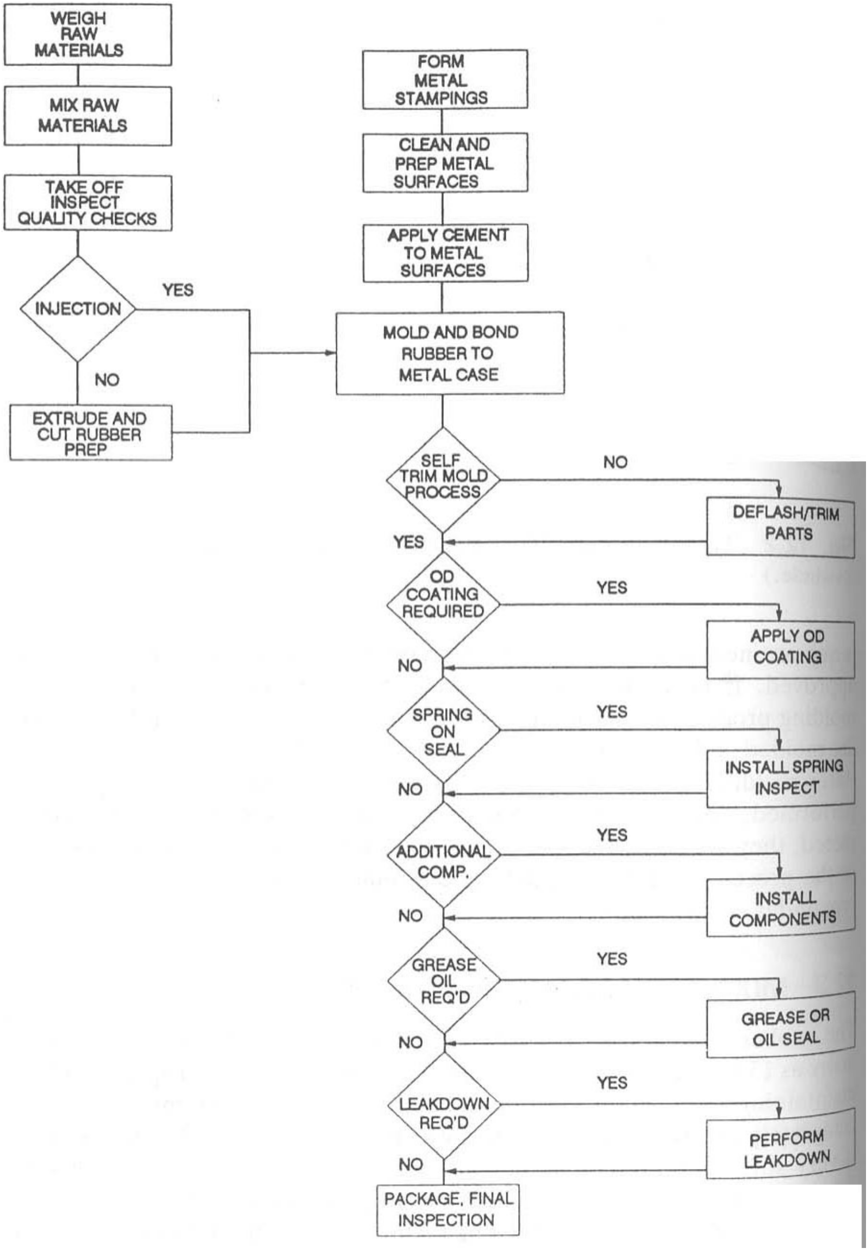

WEIGH RAW MATERIALS
MIX RAW MATERIALS
TAKE OFF INSPECT QUALITY CHECKS
INJECTION
YES
NO
EXTRUDE AND CUT RUBBER PREP
FORM METAL STAMPINGS
CLEAN AND PREP METAL SURFACES
APPLY CEMENT TO METAL SURFACES
MOLD AND BOND RUBBER TO METAL CASE
SELF TRIM MOLD PROCESS
NO
DEFLASH/TRIM PARTS
YES
OD COATING REQUIRED
YES
APPLY OD COATING
NO
SPRING ON SEAL
YES
INSTALL SPRING INSPECT
NO
ADDITIONAL COMP.
YES
INSTALL COMPONENTS
NO
GREASE OIL REQ'D
YES
GREASE OR OIL SEAL
NO
LEAKDOWN REQ'D
YES
PERFORM LEAKDOWN
NO
PACKAGE, FINAL INSPECTION

Fig. 12.3 Process flow to manufacture elastomeric seals. (Courtesy of Chicago Rawhide.)

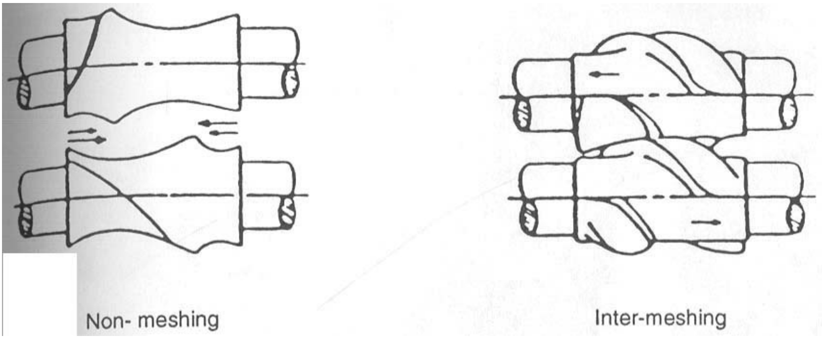


Fig. 12.4 Rotor design for mixing elastomers.

exert a high localized shear stress that will heat the material. The rotors are also designed to give low-shear stirring that homogenizes the mass.

Internal mixers have two basic rotor designs. Nonintermeshing rotor blades depend upon close clearances to provide the shear that plasticizes the mass. Intermeshing rotors have blades that mesh much like those of an egg beater (Fig. 12.4). This design provides better heat transfer than nonmeshing rotors, and is preferred for heat-sensitive materials with long mix cycles. A typical internal mixer is shown in Fig. 12.5.

The mixing cycle begins with the weighing and staging of the raw ingredients (Fig. 12.6). Many powders will absorb water, which can affect mixing, scorch, and subsequent processing. It is recommended that raw materials be stored and weighed in an environment that has humidity and temperature controls. Modern methods use bar-coded packages for the ingredients. A scanner reads the bar code and compares the code to the recipe to ensure that the correct ingredient has been chosen. If the ingredient is the right one, the computer gives the operator the green light and the material is placed upon an electronic scale. The computer releases the operator to go to the next step if the weight of material on the scale falls within the tolerance allocated for that ingredient. The process is repeated for each component of the compound. After weighing is complete, the material is staged by batch in the front of the mixer (Fig. 12.7). The batch weight depends upon the volume of the mixer chamber and the specific gravity of the compound. Typical batch weights for shaft seal compounds range from 18 kg (40 lb) to 180 kg (400 lb). Since the bag lots and weights of the ingredients have been stored in the computer, complete material traceability is ensured.

The material is then placed into the feed hopper, and a piston or ram pushes the ingredients into the rotor area (Fig. 12.8). The order of the

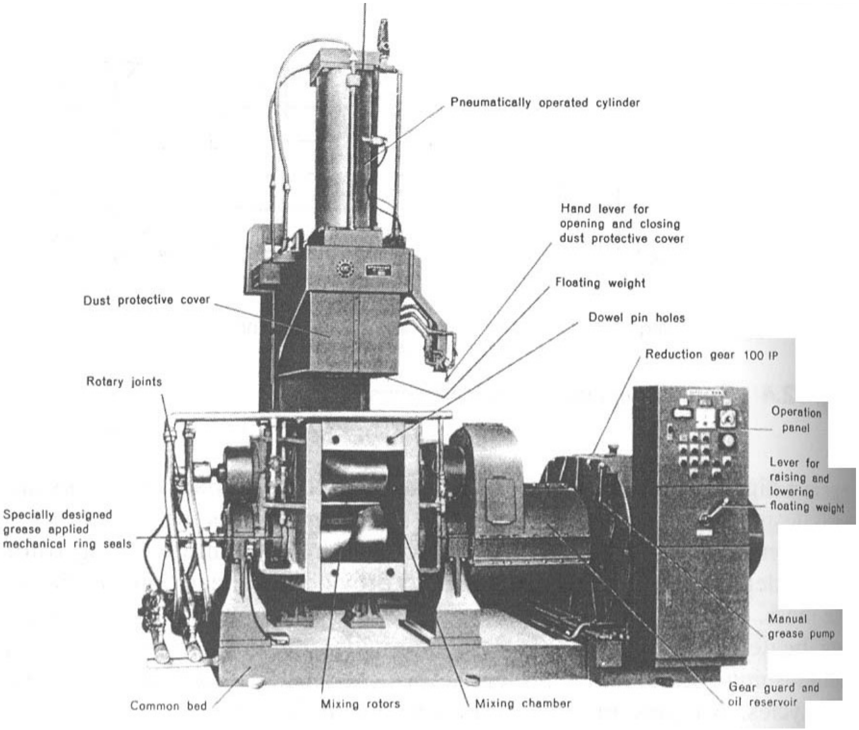

Pneumatically operated cylinder
Hand lever for opening and closing dust protective cover
Floating weight
Dowel pin holes
Dust protective cover
Reduction gear 100 IP
Rotary joints
Operation panel
Lever for raising and lowering floating weight
Specially designed grease applied mechanical ring seals
Manual grease pump
Gear guard and oil reservoir
Common bed
Mixing rotors
Mixing chamber

Fig. 12.5 Construction of a typical internal mixer.

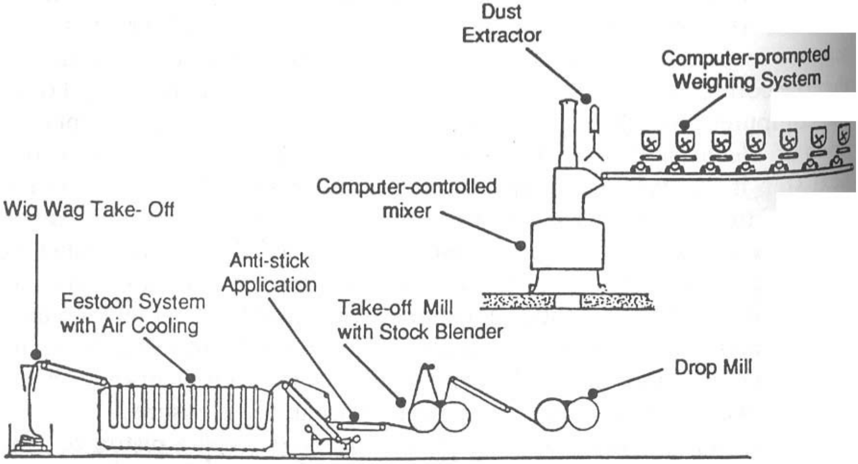

Dust Extractor
Computer-prompted Weighing System
Computer-controlled mixer
Wig Wag Take- Off
Anti-stick Application
Festoon System with Air Cooling
Take-off Mill with Stock Blender
Drop Mill

Fig. 12.6 Flow process for modern mixing line. (Courtesy of Chicago Rawhide.)

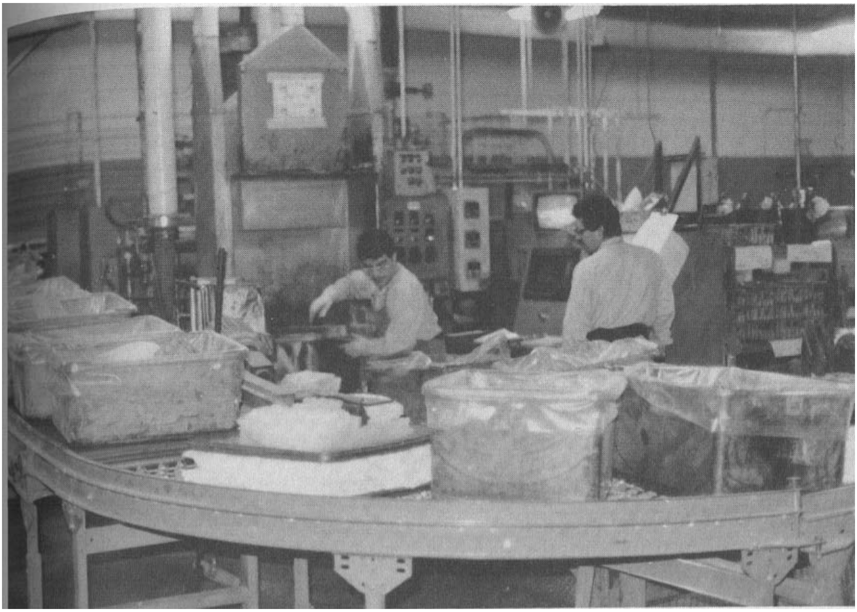


Fig. 12.7 Weighing and staging of raw materials at the internal mixer. (Courtesy of Chicago Rawhide.)

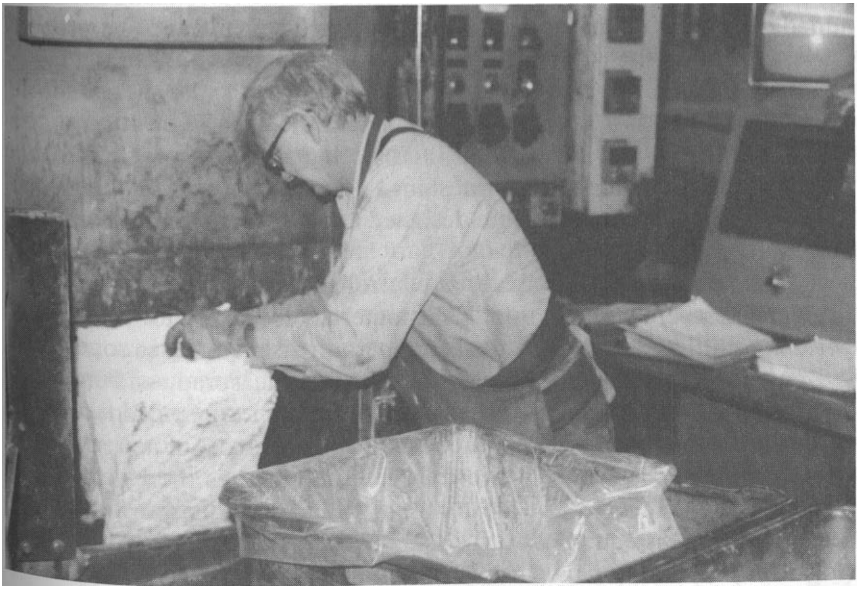


Fig. 12.8 Loading raw materials into the internal mixer. (Courtesy of Chicago Rawhide.)

addition of ingredients is important to obtain a well-mixed compound without adding too much heat. Seal compounds are thermosetting materials that will vulcanize when heated. The curatives are added during the mixing cycle, and excessive heat can cause prevulcanization or scorching, which will destroy the batch of rubber. The conventional order of addition is to add the elastomer first, lower the ram, masticate the mixture to liquefy the mass, raise the ram, and then add the dry ingredients. The ram is lowered, and more mastication takes place. Oils, plasticizers, and other liquids are added last. Fillers with very small particle sizes can be easily dispersed. The mixing time is usually long because the liquids are more difficult to incorporate once the dry fillers have been dispersed.

The fastest method of mixing is known as the upside-down method because the dry ingredients and liquids are added first and the elastomer is placed on top. All ingredients are added at once, and the ram is lowered only once. This method is ideal for those materials that have a large volume of liquids and fillers with large particle sizes. The conventional order of addition is usually used with shaft seal compounds because they are fairly dry materials and use small-particle fillers.

When the material is dry and the elastomer is difficult to plasticize, the mix may reach a high temperature that will result in premature vulcanization and scorch before the ingredients have been properly dispersed. For such materials the batch is premixed in the internal mixer without the vulcanization agents. The batch is then removed and allowed to cool. After cooling for 24 hr, it is fed back into the mixer and the vulcanizing agents are added. Two-pass mixing is required if the ingredients cannot be properly dispersed before the mix reaches a temperature of 120°C (250°F).

The rotors are driven by an electric motor. Most of the shear and frictional heating occurs between the rotors and the housing wall for a non-meshing rotor design. The bulk of the shearing action occurs between the blades of the rotors for the intermeshing design. The rotors rotate in opposite directions, and the wings are designed to cause a horizontal flow of the material.

The temperature of the mixer is controlled by water that is circulated through chambers in the housing. The materials are mixed for a given time period or until a predetermined mix temperature is reached. In some cases, the amount of energy applied to the mixer is measured, and mixing is complete when a predetermined amount of energy has been added. When mixing is complete, a door is opened and the heated mass of material drops out of the mixer and onto a two-roll mill. The mass is then worked and masticated and sheeted out. Many times the material is transferred to a second mill that is equipped with a blender mounted on the top of the mill (Fig. 12.9). The material is masticated on the mill (Fig. 12.10), and a strip

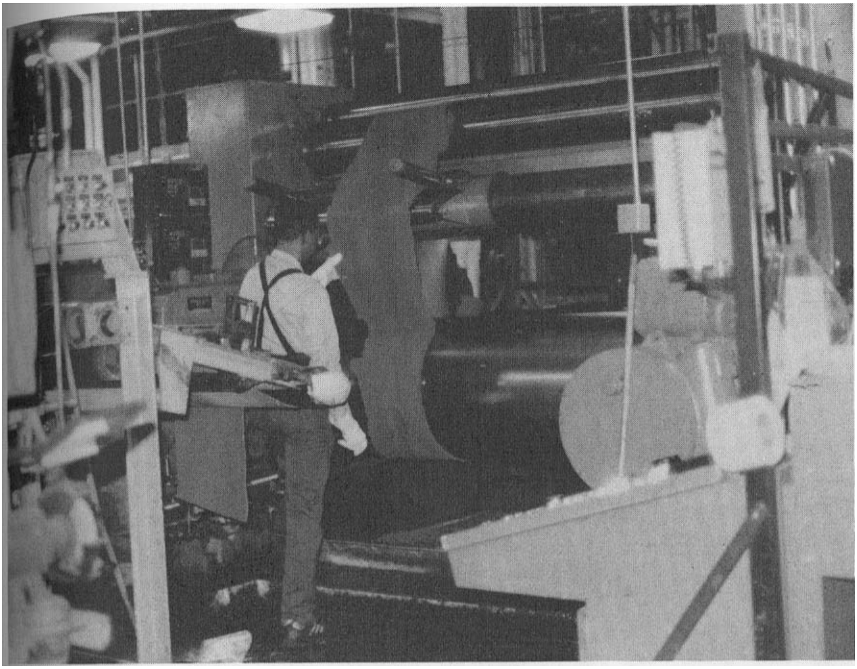


Fig. 12.9 Blending mill. (Courtesy of Chicago Rawhide.)

is fed into the blender. The blender moves back and forth in the horizontal direction, which automatically blends the material more thoroughly. Modern mixing and blending systems use computers to monitor and control the rotor speed, ram pressure, and system temperature. This control ensures consistency and quality.

After blending is complete, a strip is fed through a quick stearate dip (Fig. 12.11) to prevent the material from sticking and then into a dryer. Some materials are sensitive to water, and air drying is recommended (Fig. 12.12). After the material is dried, it is packaged for shipment to the molding area. Samples of the material are taken, and quality checks are made and compared to a statistical base. Typical measurements include hardness, elongation, and a measure of the curing characteristics using a Mooney viscometer or a rheometer. When the quality tests are complete and the material has been approved, the material is released for production. Typical quality problems and possible causes are listed in Table 12.1. The mixed compounds should be shipped and stored in climate-controlled trucks and coolers to prevent premature vulcanization.

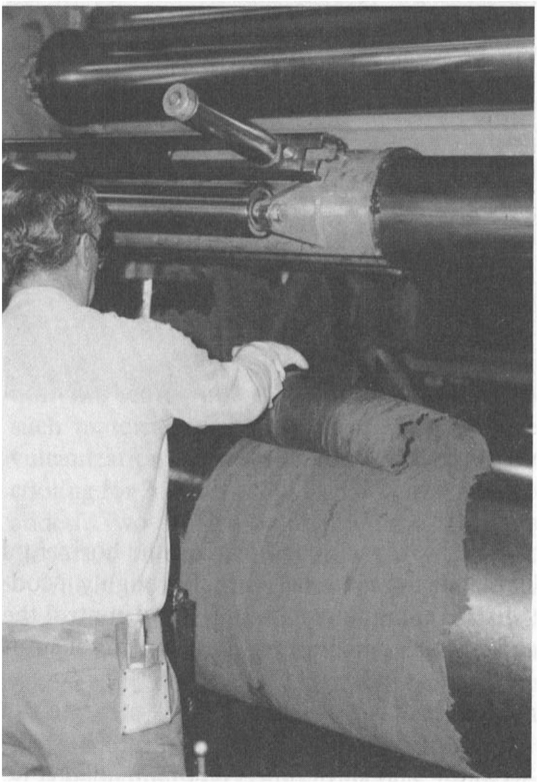


Fig. 12.10 Masticating rubber material on the blending mill. (Courtesy of Chicago Rawhide.)

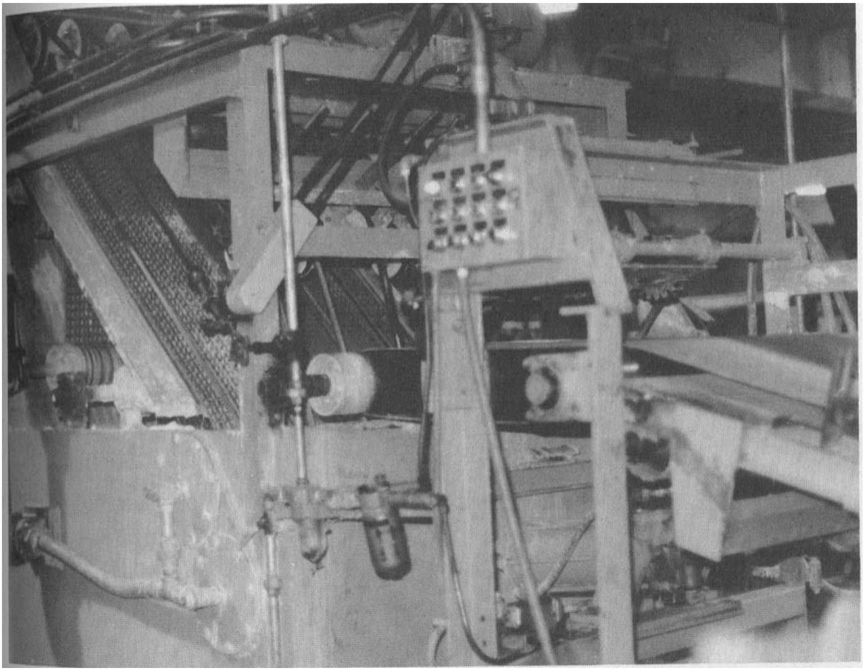


Fig. 12.11 Take-off equipment and stearate dip to prevent material sticking. (Courtesy of Chicago Rawhide.)

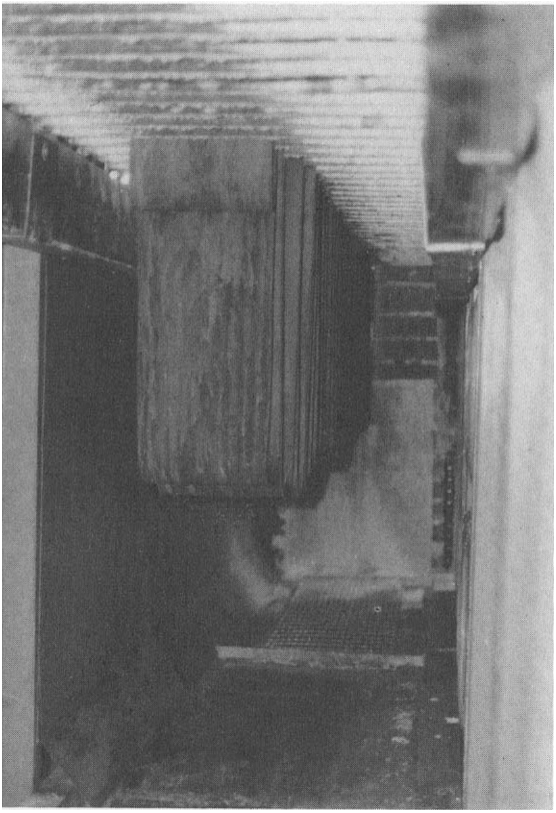


Fig. 12.12 Air drying mixed rubber material. (Courtesy of Chicago Rawhide.)

Table 12.1 Mixed Compound Quality Problems

Quality problem	Potential causes
Poor dispersion of Ingredients	1. Poor control of mixer (rotor speed, ram pressure, temperature) 2. Excessive wear of rotor or mixer walls (too much clearance) 3. Improper mixing procedure or failure to follow procedures 4. Insufficient mixing time 5. Dump temperature too high or too low 6. Improper order of addition 7. Batch size too large or too small 8. Elastomer too old 9. Excessive moisture in fillers 10. Incorrect weighing of ingredients
Scorch (prevulcanization)	1. Poor control of mixture (rotor speed, ram pressure, temperature) 2. Curing system too active 3. One-pass mix instead of two-pass 4. Dump temperature too high 5. Improper order of addition (vulcanates added too soon) 6. Initial loading temperatures too high 7. Rotor clearances too tight 8. Incorrect weighing of ingredients 9. Improper mixing procedure or failure to follow procedures 10. Improper cooling storage after mixing
Contamination	1. Poor handling and housekeeping (dust, dirt, grit, etc.) 2. Same scoop used for different ingredients without cleaning 3. Faulty rotor seals allows oil into chamber 4. Unwanted compound sticking to dump chute, mill pan, and blender 5. Unwanted compound and fillers sticking to ram, hopper, rotors, and chamber walls 6. Wrong ingredients used 7. Residue buildup at dust rings
Excessive batch-to-batch variation	1. Too much variation in rotor speed, ram pressure, and temperature 2. Errors in weighing ingredients 3. Failure to follow order of addition 4. Excessive variation in dump time and temperature 5. Variation in initial loading temperature 6. Excessive variation in raw materials

12.4 EXTRUSION OF RUBBER PREPS

After mixing, the rubber is typically sheeted off the mill in long strips. If the injection molding process is used, these strips can often be fed directly into the machine with no further work required. When the compression or transfer process is used, the strips must be processed into pills or preps that will fit properly into the mold cavity. An extruder is used to shape the rubber by forcing the rubber through a nozzle or die. Some machines use a ram to force the rubber through a die (Fig. 12.13). The strips of rubber are warmed on a small two-roll mill, and a cylindrical drum of rubber (or pig) is formed that is loaded into the extruder through a side door or into the front of the machine. The ram is activated, and the rubber is forced through a die that forms the desired shape. A cutter is attached to the front of the extruder and cuts the preps to the proper weight. The material will often swell as it leaves the die, and the die designer must consider these dimensional changes to ensure that the pill will fit into the mold. The prep weight is controlled by adjusting the ram and cutter speeds. The material weight is very important, and samples should be weighed periodically to ensure that weight control is within specifications. If the weight is too large, then the

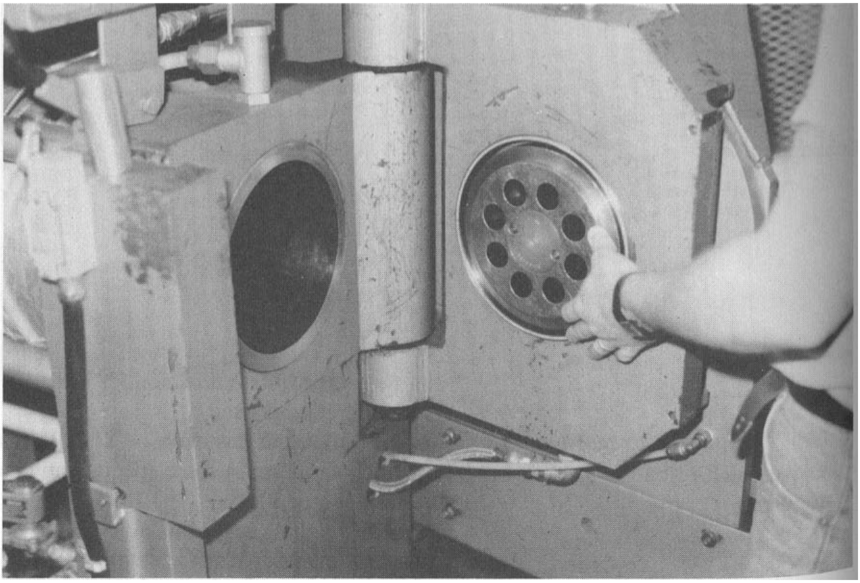


Fig. 12.13 Ram extruder. (Courtesy of Chicago Rawhide.)

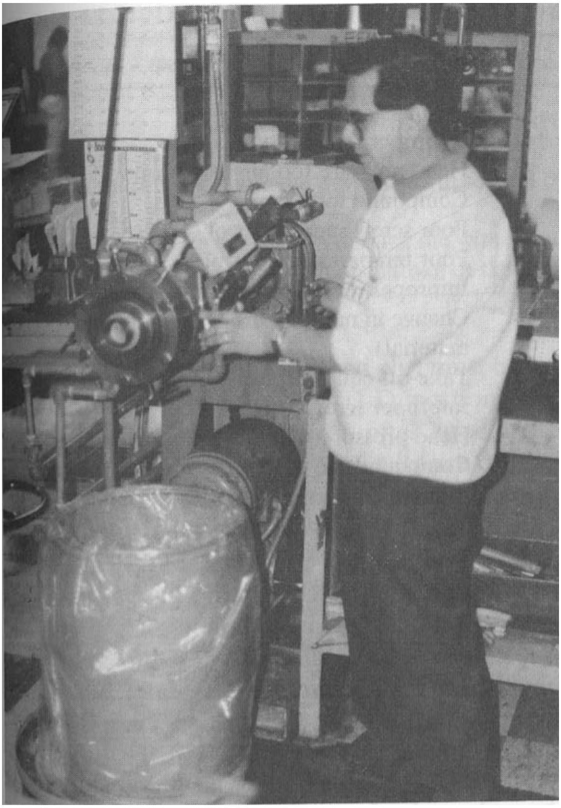


Fig. 12.14 Screw extruder. (Courtesy of Chicago Rawhide.)

mold may not close properly, excessive flash will result, and part dimensions may not meet specifications. In extreme cases, excessive weight may cause the mold to break. If the weight is too low, the mold will not fill properly, mold pressure will be too low, and poor parts will result.

A screw-type extruder can also be used (Fig. 12.14). The rubber strips can be fed directly into the barrel of a cold-fed extruder. The screw masticates the material, heats it, and forces it forward through the die, where the extrudate is cut and controlled in the same fashion as in the Ram extruder. With a hot-feed extruder, the rubber is heated before it is fed into the machine. The extrudate is cut and controlled as before. Any extrusion process will heat the rubber material. Most machines have control zones that will heat the material as it enters and cool it as it moves toward the

Table 12.2 Common Extrudate Problems

Problem	Possible cause
Low output rate	1. Wrong screw design 2. Improper temperature profile 3. Head pressure too high (improper die design) 4. Compound too scorchy
Poor dimensional stability	1. Poor screw speed control 2. Poor temperature control 3. Improper feed mechanism to extruder 4. Change in material properties (inconsistent material) 5. Take-off equipment distorts extrudate
Excessive heat generation	1. Improper screw design 2. Head pressure too high (improper die design) 3. Compound too scorchy 4. Improper temperature profile 5. Not enough cooling capacity
Rough extrudate (surface is rough, bumpy)	1. Improper screw design 2. Improper screw speed 3. Die length too short 4. Inside die surface too rough 5. Improper temperature profile

nozzle or die. The rubber prep is usually shaped like a tubular doughnut with a specified I.D., O.D., and weight. In some cases, the prep may be a cylindrical disk with only the O.D. and weight specified. After extrusion, the prep is usually hot and must be cooled to prevent scorch. It is recommended that the material be cooled by air flow through a cooling tower. Elastomeric preps should not be immersed in water. Common extrusion problems are listed in Table 12.2.

12.5 CASE MANUFACTURE AND PREPARATION FOR BONDING

The high-volume metallic cases used as components for shaft seals are usually stamped or formed out of strip steel stock that is fed into a stamping press. The strips can range in width from 25 mm (1 in.) to about 360 mm (14 in.). The thickness of the strip can range from 0.80 mm (0.0315 in.) to 3.175 mm (0.125 in.). The press force used to form the stampings can range from 54.5 metric tons (60 U.S. tons) to 364 metric tons (400 U.S. tons). The dies used to form the metal cases have various functions (Table 12.3). After

Table 12.3 Dies Used to Form Seal Cases

Primary die.	A single die that performs the first operation in one stroke of the press
Secondary die.	A single die that performs a second (or subsequent) operation to finish the part with one stroke of the press
Compound die.	A single die that can form several operations with one stroke of the press
Progressive die (prog die).	A complex die with two or more stations that perform two or more operations with one stroke of the press

the part is stamped, the finished part must be separated from the trim rings (O.D. portion) and the slugs (I.D. portion). Sometimes this is done at the press, but a tumbler sorter is often used. After forming, the metals are covered with drawing oils. The oils are often removed and the cases are covered with a rust preventive to protect the case during handling, storage, and shipping. Key dimensions (I.D., O.D., height, thickness) are statistically monitored during the manufacturing process to keep the process under control and produce parts with consistency.

Low-volume, large-diameter metal components are usually not made in stamping presses. Large flat circular blanks are fastened to the head of a lathe and rotated. The blank is then shaped with forming tools. The O.D. may be machined further to give the proper dimensions. The part is removed, and the I.D. is cut to the proper size with a circle cutter. This process is known as metal spinning, and it is more expensive than forming cases in a stamping press. Spinning is ideally suited for large-diameter, low-volume parts where the cost of a stamping die and press is prohibitive. The formed metal cases should be stored in a dry area to prevent rust.

If rubber is to be bonded to the case, then the case must be coated with cement. The metallic surface must be prepared to accept the cement and provide good adherence of the cement to the metal. Low-volume mild steel and stainless steel cases are often grit-blasted to provide a rough surface. This process can cause the case outer diameter to increase. Metal O.D. seal cases may require machining after blasting to bring the O.D. within specifications. This is not required if the seal has a rubber O.D. because the dimensions are molded. High-volume mild steel cases are usually coated with zinc phosphate, which provides a rough surface and also protects the case from rust. The modern process consists of a series of dip tanks with a rotating basket that travels on a track. The cases are loaded into a basket, and the basket moves down the track to the appropriate tank. It is lowered into the tank and then rotated to ensure that every portion of every case is exposed to the solution. The sequence, time in the individual tank, and rate

Table 12.4 Typical Sequence to Apply Cement to Seal Cases

1. Alkaline stripper tank	Strip parts of oil.
2. Alkaline stripper rinse	Rinse parts.
3. Alkaline cleaner tank	Clean parts.
4. Alkaline rinse tank	Rinse parts.
5. Acid pickle tank	Remove oxides from surface.
6. Acid rinse parts	Rinse parts.
7. Zinc phosphate tank	Apply phosphate.
8. Phosphate rinse	Rinse parts.
9. Waterbase cement tank	Apply water-based cement.
10. Dryer	Dry parts.

of basket rotation are controlled by computer. Some materials can be bonded to the case with an environmentally safe water-based cement. This can be placed in line with the phosphating process. A typical sequence is described in Table 12.4. Wastewater from the system must be treated before being released into the environment.

Many compounds will not bond to the mild metal case with water-based cement. Solvent cements are used to bond these compounds to mild steel, stainless steel, aluminum, and brass cases. They are usually sprayed onto the metals, and the overspray must be captured with scrubbers before the fumes are released into the atmosphere. The cemented stampings should be taken to the press line as quickly as possible for bonding. The maximum shelf life of the cement is 3 days.

12.6 MOLDING TECHNIQUES

12.6.1 Mold Design

The molding process actually begins with the design of the mold itself. The dimensions of the mold components form the cavity that will contain the metal and outline the seal cross section (Fig. 12.15). Elastomers create high pressure during the molding process. The rubber must be vulcanized or cured, so heat must be added. Some elastomers have ingredients that may cause the rubber to stick to the mold or dirty the mold surface. Dirty molds will result in seal defects.

The molds must be made to exact dimensions, and if a multicavity mold is used, the cavities must be identical. The molds are typically made of a hardened tool steel that will resist handling damage and withstand the high internal pressures generated during the molding process. The surface of the

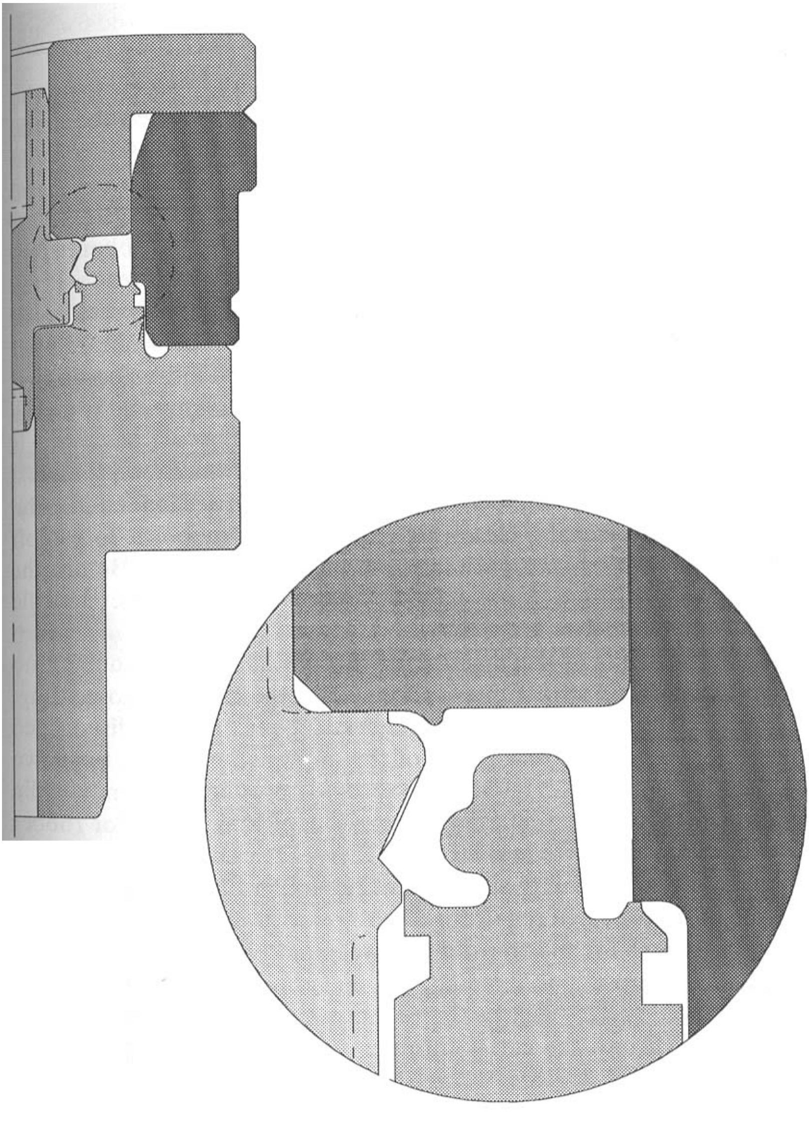


Fig. 12.15 Mold dimensions form the cavity that contains the seal metal and defines the seal cross section. (Courtesy of Chicago Rawhide.)

mold insert is often chrome-plated to prevent rust and reduce mold fouling. Many key dimensions are ground to provide the necessary accuracy.

The mold design is highly dependent upon the molding process and the product design. The compression molding process begins when the metal and rubber prep are placed in the mold and the mold is closed. As the mold closes, the rubber is squeezed into the cavity, where it bonds to the metal and vulcanizes. If the part has a metal O.D., then the mold design must include a feature to close the cavity and prevent the escape of rubber before the mold is fully closed. This feature is known as closed mold stroke. If the cavity is not closed properly and rubber flows into other areas of the cavity and along the metal stamping, then extra operations will be required after molding to remove the flash. Also, pressure may not build up to desired levels, and molding defects may occur. The rubber may not bond properly to the metal. A mold design that provides the desired closed mold stroke is illustrated in Fig. 12.16. Two components of the mold are spring-loaded with Bellville washers. The rubber prep and metal shell are loaded into the mold and the mold begins to close (Fig. 12.16a). The rubber is forced off the shelf into the cavity as the mold is closing (Fig. 12.16b). Before the mold is fully closed the outer mold ring loaded with the large Bellville washers will contact the stamping and seal the cavity. The inner ring, which is loaded with the small Bellville washers, will continue to push rubber into the cavity and complete the filling process after the mold is closed (Fig. 12.16c). When the mold is opened, the excess rubber or cap is disengaged from the seal and remains on the top of the mold, where it can be blown away with an air hose (Fig. 12.16d). The resulting product has a molded lip and requires no additional deflashing operations. Mold design for rubber O.D. seals may not require springs (Fig. 12.17). A closed mold stroke is not required to seal the cavity and prevent rubber flashing along the metal because the product design requires rubber to flow to the O.D.

The mold designs for the transfer and injection processes are basically the same as for compression molding, but no springs are required to provide a closed mold stroke because the rubber is forced into a closed cavity. Rubber is injected into the cavity through a central vertical sprue and a horizontal runner (Fig. 12.18).

Different mold designs can result in added operations after the part is vulcanized. If the gates are thin enough, the unwanted flash will tear when the mold is opened and remain on the top, where it can be blown away with an air hose (Fig. 12.19a). In some cases, the flash tab is too thick and must be trimmed. In Fig. 12.19b, the O.D. flash will be torn away and remain with the top when the mold opens. The flash must be trimmed away from the seal head, but the lip dimension is still molded. Some mold designs require trimming of both the O.D. and the I.D. The seal contact point is formed by the trim knife (Fig. 12.19c).

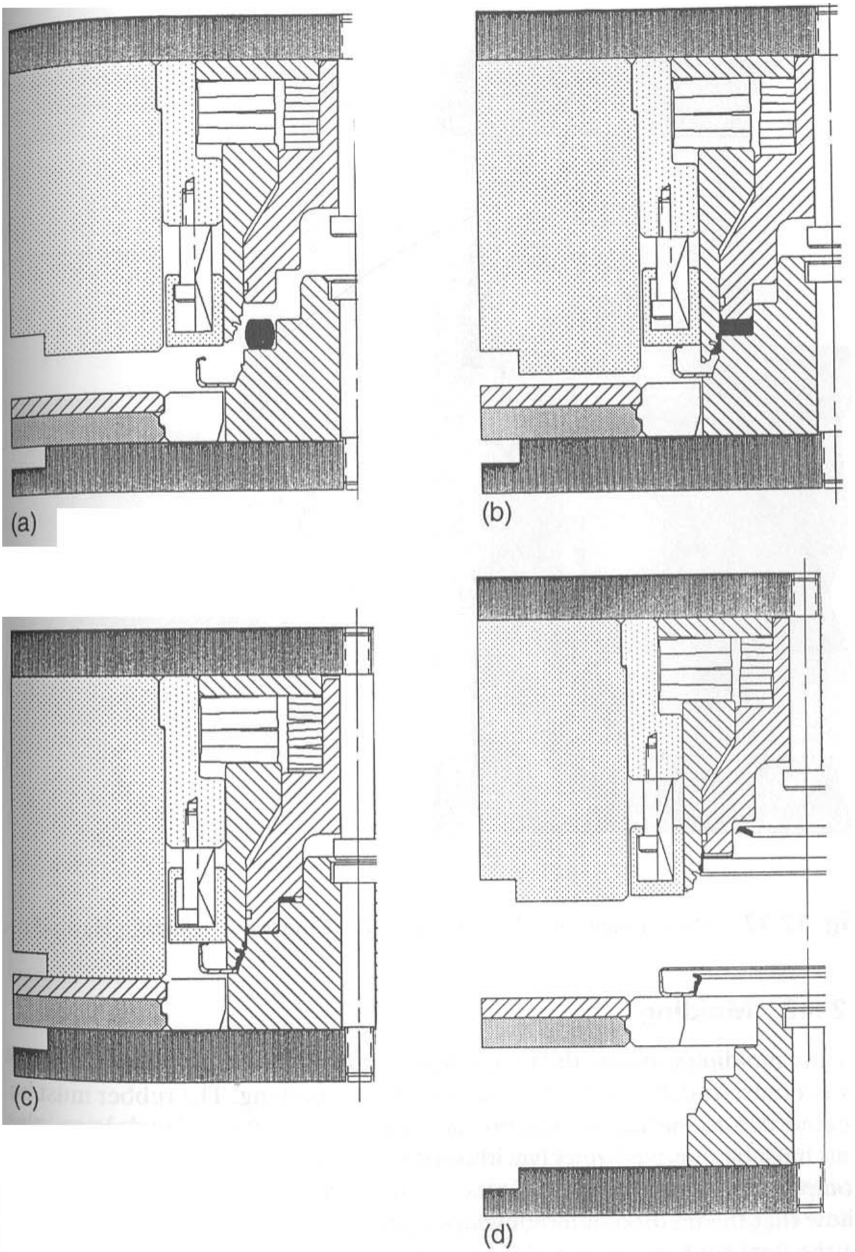


Fig. 12.16 Compression mold with closed mold stroke for metal O.D. seals. (a) Prep and metal loaded; (b) mold closing; (c) mold closed; (d) mold open. (Courtesy of Chicago Rawhide.)

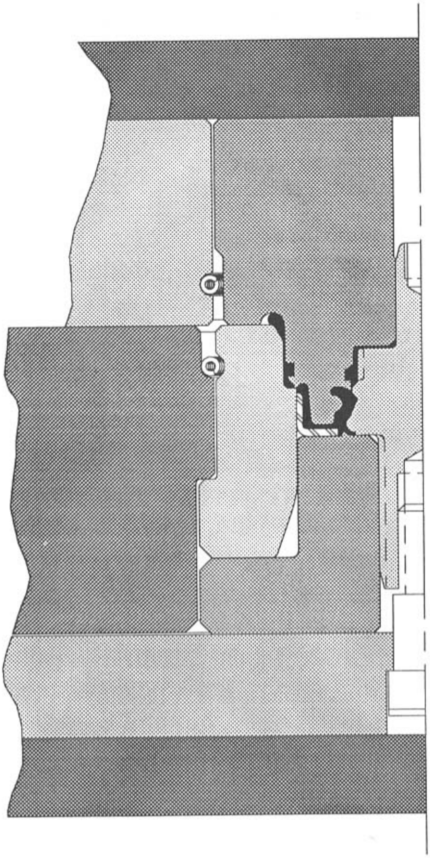


Fig. 12.17 Mold design for rubber O.D. seals. (Courtesy of Chicago Rawhide.)

12.6.2 Molding

In any molding process, the metal must be placed in the mold and the elastomer must be delivered to the cavity without scorching. The rubber must be heated before curing or vulcanization can occur. After vulcanization, the part must be removed quickly without damage. Three molding techniques--compression, transfer, and injection--are used to make oil seals. Studies show that the method of molding does not affect the mechanical properties of the final rubber product [34].

The mold consists of one or more cavities that are fitted into plates. The upper plate will form the upper portion of the molded part and is bolted to the top platen of the molding press. The bottom plate forms the lower

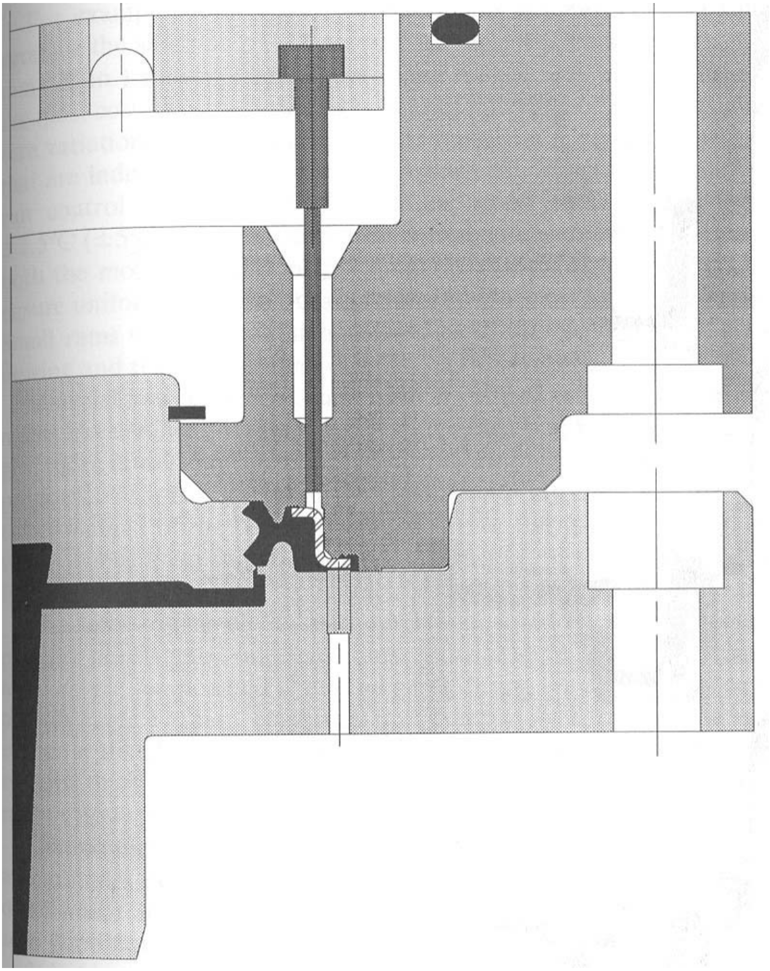


Fig. 12.18 Injection mold design. (Courtesy of Chicago Rawhide.)

portion of the product and is bolted to the bottom platen in the press. The press platens contain the source of heat. Old methods used steam or hot oil, which resulted in leaks, poor response, and fairly poor temperature capabilities. Modern methods use resistance or induction electric heating platens, which can provide rapid heating response and mold temperatures up to 204°C (400°F). Hotter molds result in quicker part vulcanization, faster cycles, and lower costs. The variation in temperature across the top and bottom halves of the mold is a function of the press platen size, proper fit of

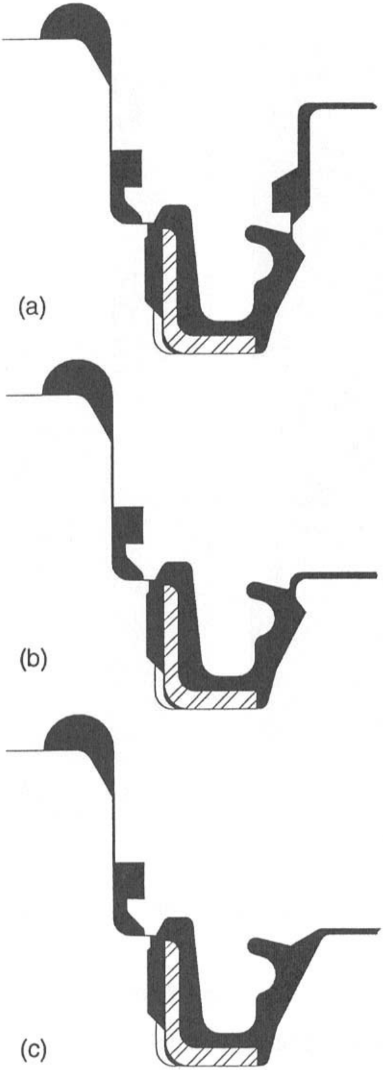


Fig. 12.19 Mold design can result in added operations. (a) Flash removed when mold opens. No added operations. (b) O.D. flash removed when mold opens. I.D. must be trimmed with molded lip seal. (c) O.D. flash removed when mold opens, ID is trimmed. (Courtesy of Chicago Rawhide.)

the mold and press plate at their interface, mold construction, mold open time, and heating system design. It is more difficult to maintain large platens at constant temperatures. One solution is to use presses with small platens.

For example, four presses with a platen 355 mm (14 in.) square will produce the same output of product as one press with a platen that is 710 mm (28 in.) square. Careful economic studies are required to determine the most cost-effective press size. Another method of minimizing temperature variation across the mold is to use multizone electrical heating systems that are independently controlled. A correctly balanced multizone system can control the temperature variation across a large platen to within ±2.5°C (±5°F). The pressure distribution across the mold is a function of both the mold and press platen flatness. The two must mesh properly to ensure uniform pressure. A large ram will also provide a uniform pressure. Small rams will create a higher pressure in the center, which may create bowing and result in product dimensional variations.

Material flow in the mold can result in poor oil seal performance. Experiments have shown that seals made with molds that have shelf-loaded preps give more consistent functional performance than those with cavity-loaded preps [9]. This is also true if the elastomer is injected into the top of the cavity instead of the bottom. The reason for this is not completely understood. It has been suggested that the direction of flow aligns the fillers of the compound in a preferred direction that enhances performance.

The appearance of the part improves, molding defects are reduced, product consistency improves, and functional reliability increases if the mold cavity is evacuated of air and gases during the molding process. Some presses have vacuum chambers that completely surround the mold and evacuate the chamber and the mold cavities before they are closed. Other systems draw a vacuum when the mold cavities are closed just before the rubber flows into the cavity.

Compounds used in the injection process must flow readily in the sprues and runners before they enter the cavity. The rubber compounder must often make adjustments to the recipes of compression compounds to ensure good flow.

It is essential that the elastomeric compound reach the desired cure state in the shortest possible time to maximize productivity while maintaining product integrity. Most modern presses use microprocessor controls that can adjust press conditions to accommodate changes in platen temperatures and compound processing characteristics without compromising product performance.

Compression presses are produced in various sizes throughout the world. The type and size of press are dependent upon product type, product size, and product volume. In general, large deck presses are the most economical choice for large-volume production. Smaller presses are preferred for low volumes. The press tonnage is also important to ensure that the press is properly closed with the desired cavity pressure. As the press platens get larger, more load is required. The load should be approximately

800,000kg/m^2 (1130 psi). The molding process is fairly simple. The metal and rubber preps are loaded into the molds by hand if the number of cavities in the mold is small. If there are a large number of cavities (usually over nine), then loading trays may be used. The operator will place the metals and preps into trays while the press is closed and the previous load is being vulcanized. When the press opens, the operator will remove the finished parts, blow away flash with an air hose, spray the mold release if required, and then place the tray full of metal cases over the mold. The operator will trip a release that allows the metal cases to drop into their proper position in the mold. The process is repeated with the tray that contains the rubber prep. Sometimes both the metal cases and the rubber prep can be combined into a single tray. The mold is closed, and the process is repeated. The mold may be opened slightly (1--2 mm) for a fraction of a second several times after initial close to help drive air and gases out. This process is known as bumping. Cycle time will vary (from 2 to 10 min) and depends upon the compound cure cycle and the time required to load the trays. A typical compression press is shown in Fig. 12.20.

Injection molding eliminates the need to make rubber preps that must be loaded into the mold. Rubber strips are fed into the machine, warmed, and injected into a closed cavity. Because the rubber is warmed, cure times and cycle times can be reduced. Three basic types of machines are used to injection mold shaft seals. The horizontal reciprocating screw machine with a vertical platen is the least used because it is more difficult to load and hold metal inserts than to operate the injection molding machine with a vertical screw and a horizontal platen. A second type of injection machine has a screw and a ram. The rubber strip is fed into the screw, and the screw forces the rubber into the cylinder. When the cylinder is full, the ram forces the rubber into the mold cavities. The reciprocating screw machine has a direct approach and fewer components and is often preferred over the screw/ram machine. The rubber strip is drawn into the machine by the screw, which masticates it and forces it into a cylindrical area just in front of the screw. When the cylinder is completely filled, the screw is rammed forward and the elastomer is forced into the mold cavities.

All presses must have adequate controls for controlling the temperature in the injection head and the mold, the injection pressure, and the volume (shot size) of material that is injected into the cavity.

The vertical presses can be modified by adding a rotary table to increase output. A rotary table with eight stations would have one mold at each station. The table would rotate under the injection head, and material would be injected at the first station. The next six stations would allow the parts to cure, and the eighth station would be the open, unload, and load point. The vertical machines are easier to automate for the load/unload procedure than the horizontal machines [10, 17d].

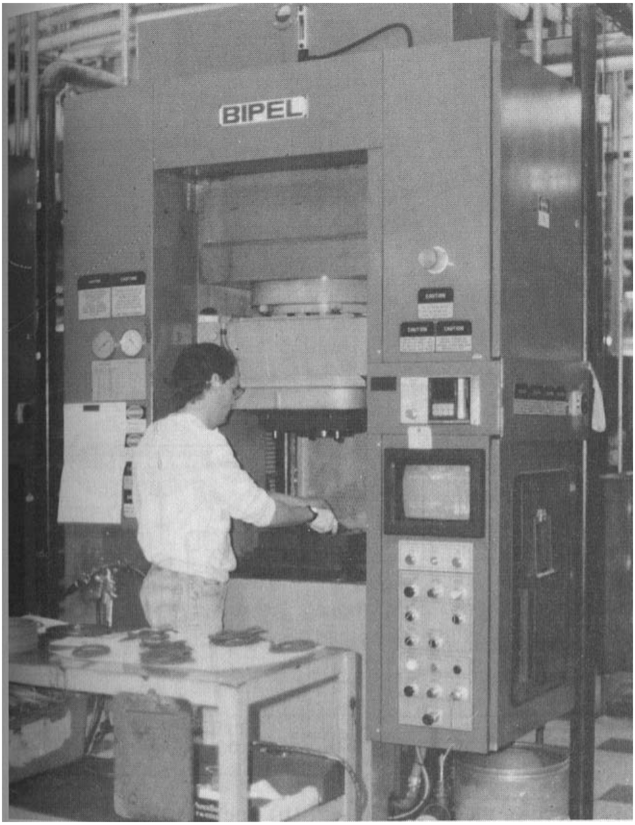


Fig. 12.20 Modern compression press with vacuum boxes. (Courtesy of Chicago Rawhide.)

If a multicavity mold is used, then the rubber must flow from the nozzle through a sprue to the runner system and then to the individual cavities. A typical four-cavity mold is shown in Fig. 12.21, and the runner system for the parts appears in Fig. 12.22. The sprue, runner system, and flash caps are all waste because they have been vulcanized along with the product. This waste can be eliminated if an injection/transfer system can be combined with cold pot technology.

Injection/transfer molding is generally used for high-volume manufacture. The transfer “pot” replaces the runner system in a conventional injection mold; it contains a ram that forces the uncured compound into the individual cavities making up the mold. The pad of rubber in the pot will vulcanize, resulting in waste that is very undesirable when using the more

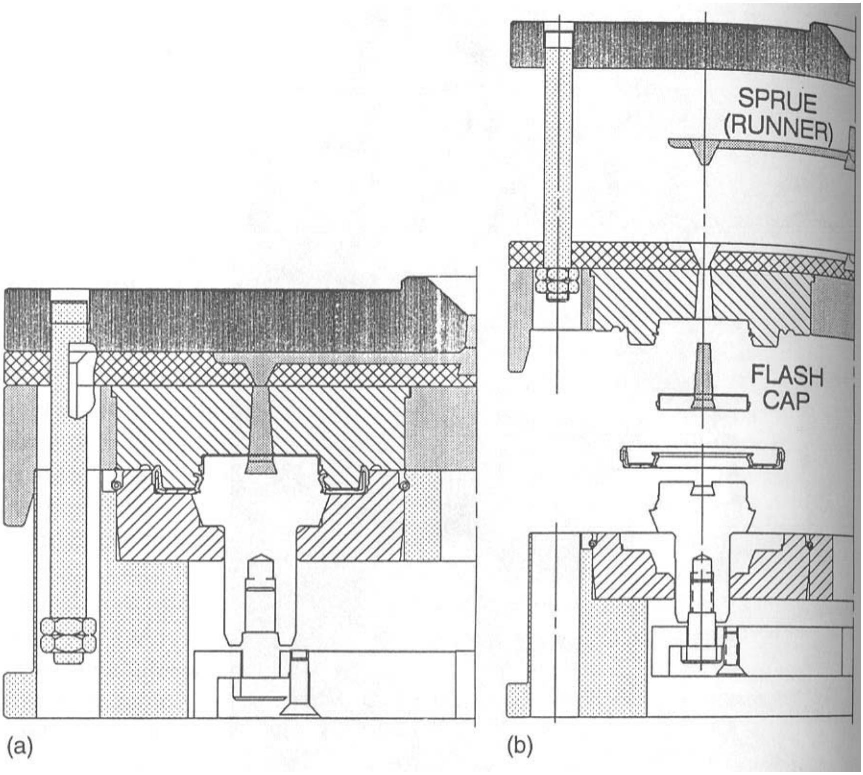


Fig. 12.21 Four-cavity injection mold. (a) Closed; (b) open. (Courtesy of Chicago Rawhide.)

expensive elastomers. To overcome the compound waste issue, one approach has been the development of the "cold pot" system, which minimizes the amount of waste compound generated during the molding process. The use of a cold pot injection/transfer molding process gives all the advantages of high-volume compression molding without having to make rubber preforms and then load the preforms into each cavity.

It is important to note that the molding cycle is different from the conventional injection process. The major difference is that when the rubber compound has been moved from the transfer pot to the individual cavities, the transfer pot is then removed and the "hot plate" is moved in to take its place. Figure 12.23 illustrates a typical mold/transfer pot layout that can be used in the injection/transfer process. The layout also shows the hot

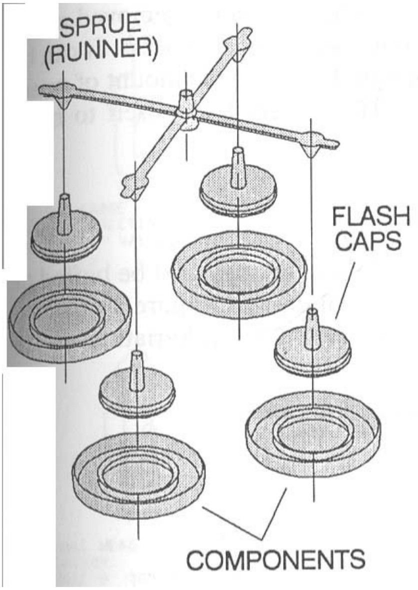

SPRUE
(RUNNER)
FLASH
CAPS
COMPONENTS

Fig. 12.22 Sprue, runner system, flash caps, and components for a four-cavity injection mold. (Courtesy of Chicago Rawhide.)

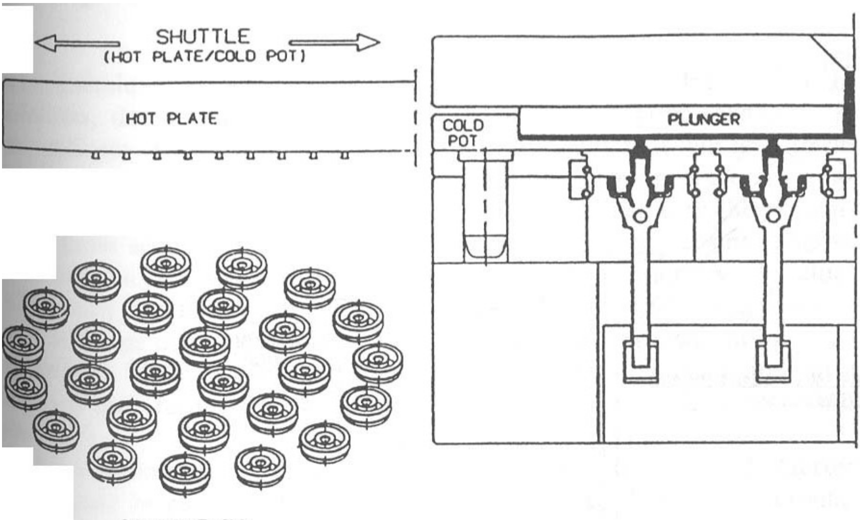

SHUTTLE
(HOT PLATE/COLD POT)
HOT PLATE
COLD
POT
PLUNGER
COMPONENTS

Fig. 12.23 Cold pot injection transfer. (Courtesy of Chicago Rawhide.)

plate ready to move into position when the transfer pot is removed and typical layout for seals produced in the injection/transfer process. It can be seen that there is no flash or waste compound. The small amount of waste is, in fact, removed with the hot plate. This system lends itself to high volume automation.

12.7 FINISHING OPERATIONS

If pick trim molding concepts are not used, then the seal must be buffed or trimmed to remove unwanted flash. Some seal designs require a trimmed lip that is formed by a trim knife and a lathe (Fig. 12.24). Variations in the

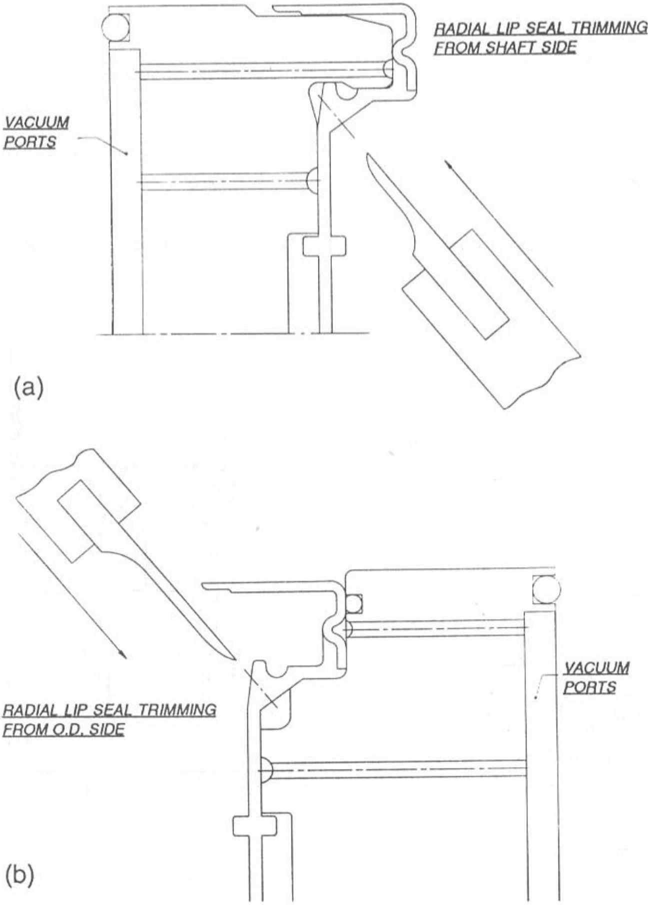

RADIAL LIP SEAL TRIMMING
FROM SHAFT SIDE
VACUUM
PORTS
(a)
RADIAL LIP SEAL TRIMMING
FROM O.D. SIDE
VACUUM
PORTS
(b)

Fig. 12.24 Trimming seal lips (a) From shaft side; (b) from O.D. side.

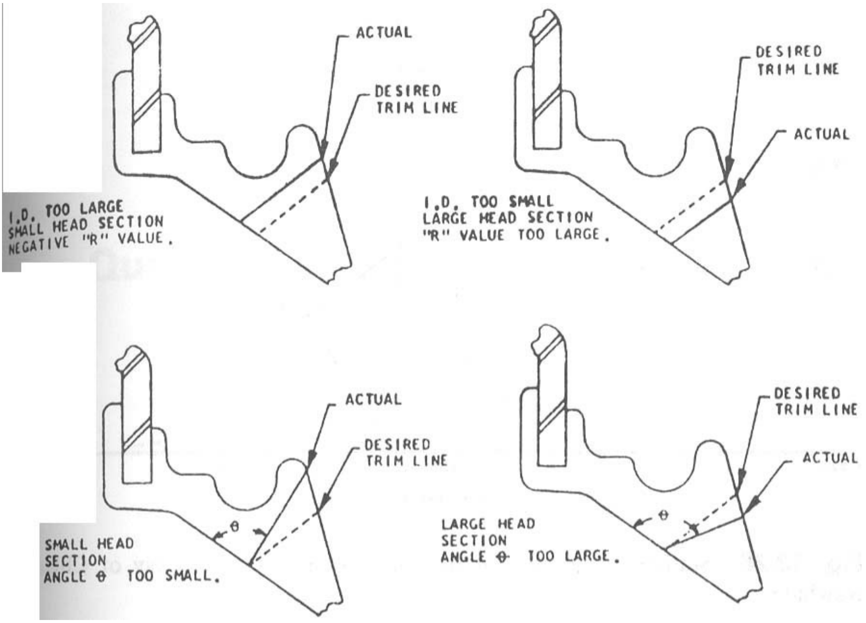

ACTUAL
DESIRED TRIM LINE
I.D. TOO LARGE
SMALL HEAD SECTION
NEGATIVE "R" VALUE.
DESIRED TRIM LINE
ACTUAL
I.D. TOO SMALL
LARGE HEAD SECTION
"R" VALUE TOO LARGE.
ACTUAL
DESIRED TRIM LINE
θ
SMALL HEAD
SECTION
ANGLE θ TOO SMALL.
DESIRED TRIM LINE
ACTUAL
θ
LARGE HEAD
SECTION
ANGLE θ TOO LARGE.

Fig. 12.25 Improper trim procedures can affect product cross section. (Courtesy of Chicago Rawhide.)

trim technique produce variation in key seal parameters (Fig. 12.25). In addition, double cuts and nicks from the knife give poor reliability. A nonuniform cross section will produce uneven loads that will lead to an uneven wear path and high local temperatures. This problem is compounded when the shaft-to-bore misalignment coincides with the nonuniform cross section. These installation variables interact, causing random seal failures. Excessive trim variation may result in a negative R value, which may lead to seal instability and leakage. Trim tolerances will produce a wide range of free lip dimensions (Fig. 12.26), which result in lip force variations. If the seal-to-shaft interference is too light, misting and early leakage may result. Excessive interference may result in high wear and short life.

After trimming is complete, other components are installed. Garter springs can be placed in the spring groove by hand or machine. If automation is used, a magnetic or visual defection device should check to ensure that only one spring is present and that it is properly installed. Other components such as inner cups, plastic backup rings, and grease can be added automatically on an in-line setup. Automatic checks should be made

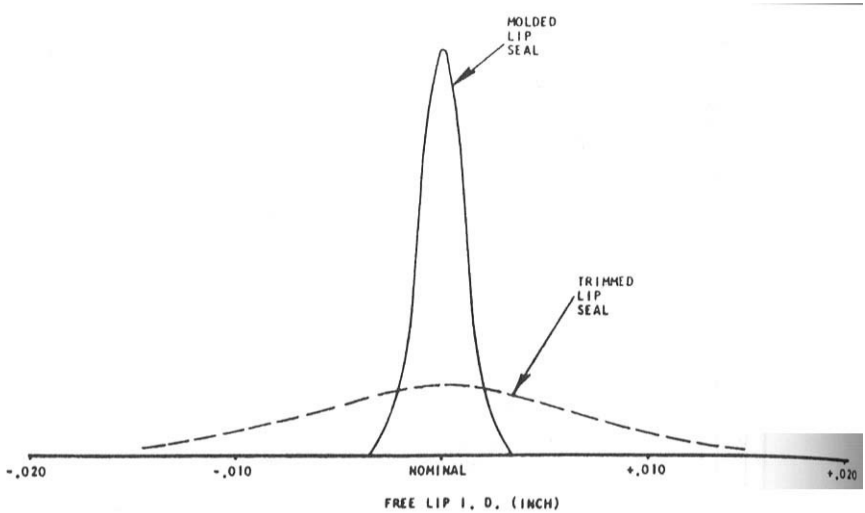


Fig. 12.26 Statistical variation of free lip dimensions. (Courtesy of Chicago Rawhide.)

after each operation to ensure that the operation was properly performed. Air leakdown and LOP can also be performed in line to ensure lip integrity. After all operations are complete, the parts are packaged to customer specifications, audited, and shipped.

13

Quality Assurance for Sealing Systems

13.1 RADIAL LIP SEAL CHARACTERISTICS

Seal users and suppliers work together to define quality systems that provide reliable products at reasonable cost [11, 39, 232]. Inadequate quality systems may result in nonconforming products that experience early field failures. Overspecifying the product and demanding needless measurements, reporting systems, and functional audits will increase costs without improving reliability. The Rubber Manufacturers Association (RMA) has identified a number of radial lip seal characteristics that have been used by seal users and suppliers to define the product [165]. Not all of these characteristics are critical to seal function, and those that are not should not be controlled in the same manner as the critical ones. The RMA has identified the characteristics that are critical and should be monitored using statistical process control (SPC) during the manufacturing process (Table 13.1). Process control records for these characteristics are maintained at the production plant site. The other characteristics are defined by supplier certification or quality audits that meet the desired statistical requirements. Some of the characteristics are defined for reference use and are not required for product definition. If the seal user requests additional controls or nonstandard tolerances for a seal characteristic, then additional costs may be assessed. In some cases, the seal user may be using measuring techniques that differ from those used by the seal supplier. The RMA has provided the

Table 13.1 Radial Lip Seal Characteristics

Characteristic Requirements	Characteristic
Critical seal characteristics (SPC required)	Primary lip sprung I.D.
	Primary lip unsprung I.D.[a]
	Secondary lip I.D.
	Seal outer diameter and geometry
	Seal width
	Spring I.D. or free lengtha
	Spring load at specified deflectiona
	Unsprung radial wall variationa
Seal characteristics certification from supplier acceptable	Elastomer-to-metal case bond
	Elastomeric material line callout
	Elastomeric material specific gravity
	Elastomeric material microhardness
	Case material
	Case thickness
	Spring material
	Spring stress relief stability
	Spring joint integrity
	Spring coil diameter
Seal characteristics quality audit (statistical conformance required)	Seal cross-sectional geometry
	Rubber flash
	Seal identification
	Proper hydrodynamic feature
	Lead in chamfer on seal toe
	Lip heights
	Proper packaging
	Proper prelube presence
	Spring presence
	Visual variations
Seal characteristics for reference only	Coating thickness on seal case
	Primary and secondary lip contact heights on shaft
	Primary and/or secondary lip torque
	Radial load or lip opening pressure
	Pressure test
	Radial wall dimension
	Sprung radial wall variationa

[a] The seal manufacturing facility controls this characteristic internally. The customer receiving inspection should not measure this characteristic.

Source: Ref. 165.

industry with referee measuring methods that should be used to settle disputes [160].

The RMA has also prepared documents that can be used to survey supplier quality systems [169-173].

The elastomeric material is critical to seal function. Incoming raw materials must be analyzed to ensure conformance to specifications [228]. It is not necessary for the user to inspect the material if the supplier has been approved to certify the raw materials. The RMA has prepared documents that describe the product characteristics of polymers and other key ingredients used to formulate oil seal compounds. These documents also contain quality control report forms for each ingredient [174-185].

Modern methods of assuring quality are followed to control the manufacturing processes and reduce variation. The process capability Cp is defined as the specification width of the seal characteristic divided by the process width and is a measure of the spread of the process. It is calculated with the equations

$$\bar{X} = \sum_{i=1}^{N} \frac{X_i}{N} \tag{13.1}$$

$$s = \left(\frac{\sum_{i=1}^{N} (X_i - \bar{X})^2}{N - 1} \right)^{1/2} \tag{13.2}$$

$$C_p = \frac{USL - LSL}{6s} \tag{13.3}$$

where is the average value of the seal characteristic, Xi is the specific value of the seal characteristic, N is the number of measurements of the seal characteristic, USL denotes the upper specification limit, and LSL denotes the lower specification limit.

Cp assumes that the process is positioned at the exact center of the seal characteristic specification. If the process capability is centered and equal to the seal characteristic tolerance, then the value of Cp is 1. In actual practice, processes are seldom if ever centered. A modification of Cp is required to account for shifts in the process average. This modification is called Cpk and uses the spread between the process average and the closest drawing specification for the numerator and half of the process capability for the denominator:

$$C_{\text{pk}} = \left| \frac{\bar{X} - \text{Nearest spec limit}}{3s} \right| \tag{13.4}$$

Table 13.2 Effect of Changes in Cp and Cpk

	Shift in Cpk (Cp - Cpk)				
Cp	0.00 (Cp = Cpk)	0.10	0.20	0.30	0.40
0.50	133,614[a]	151,000	201,925	282,451	385,556
0.80	16,395	21,331	37,280	67,291	115,229
1.00	2,700	3,950	8,357	17,913	35,944
1.10	967	1,509	2,274	8,211	17,868
1.20	318	532	842	3,470	8,198
1.30	96	172	487	1,351	3,467
1.40	27	51	160	474	1,350
1.50	7	14	48	159	483
1.60	2	4	13	48	159
1.70	0	1	3	13	48
1.80	0	0	1	3	13
2.00	0	0	0	0	1

[a] Nonconforming parts per million.

Cp and Cpk are identical if the process average lies precisely between the upper and lower specification limits. The Cpk value decreases as the process average moves toward either specification limit. The effect of reducing the process spread and centering the process is tabulated in Table 13.2. If Cp = 1.00 = Cpk (the process is centered), then there will still be 2700 nonconforming parts per million. If the process center shifts by 0.20, the number of nonconforming parts will increase to 8357 parts per million. Many seal customers require a Cpk of 1.33 and expect the seal supplier to reduce process spread and increase Cpk values to 1.66 and possibly 2.00.

As an example, the specification of the sprung inner diameter of an oil seal is 50 ± 0.5 mm. The upper specification limit (USL) is 50.5 mm, and the lower specification limit (LSL) is 49.5 mm. The average value (X) of 50 measurements is 50.143 mm with a standard deviation (s) of 0.119 mm. C_p can be calculated as

$$C_p = \frac{\text{USL} - \text{LSL}}{6s} = \frac{50.5 - 49.5}{6\ (0.119)} = \frac{1.00}{0.714} = 1.400$$

From Table 13.2, 27 seals out of a million will have a sprung inner diameter beyond the specification limits with a 1.33 Cp value. The process is not centered, because X is larger than the desired dimension. The value of Cpk can be calculated as

$$C_{pk} = \left| \frac{\bar{X} - \text{Nearest spec limit}}{3s} \right| = \left| \frac{50.143 - 50.5}{3\ (0.119)} \right| = 1.0$$

The difference between the Cpk and Cp is 0.4. The number of nonconforming seals can be found from Table 13.2 to be 1350 with Cp = 1.4 and Cp - Cpk = 0.4. The importance of centering a process is clear.

13.2 MEASURING PRIMARY AND SECONDARY LIP INTEGRITY

If the contact bands of the primary and secondary lips have areas where no contact is made with the shaft, leakage or contaminant ingression may result. Several methods are available to determine if the contact bond is broken [168].

13.2.1 Light Box Test Procedure

A test shaft or mandrel is designed to be screwed into a light box (Fig. 13.1). The mandrel has a surface finish that is less than 0.76 µm (30 µin.), and the diameter is 0.38 mm (0.015 in.) smaller than the application shaft diameter. If the shaft-to-seal designed lip interference is less than 0.76 mm (0.030 in.), then the mandrel must be made smaller than the application shaft diameter by one-half of the design interference. If the seal has a secondary lip, then an undercut is provided to ensure that the light will not be blocked (Fig. 13.2). The seal is placed over the mandrel with the lip that is to be inspected facing up, and the tester light is turned on. The seal is acceptable if no light is visible anywhere around the circumference of the seal lip and test shaft interface. This method is slow, subjective, and unsuitable for high-speed mass production. It is useful for SPC audits.

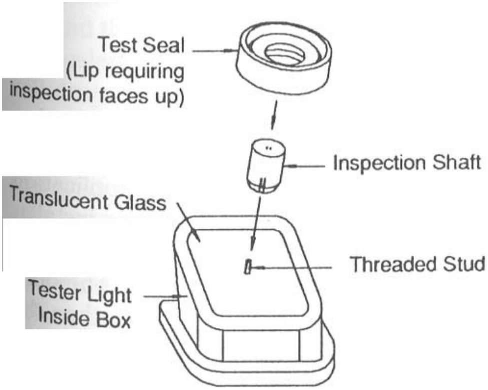


Fig. 13.1 Typical light box. (Courtesy of Rubber Manufacturers Association.)

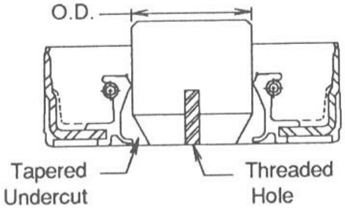

O.D.
Tapered Undercut
Threaded Hole

Fig. 13.2 Light box inspection shaft. (Courtesy of Rubber Manufacturers Association.)

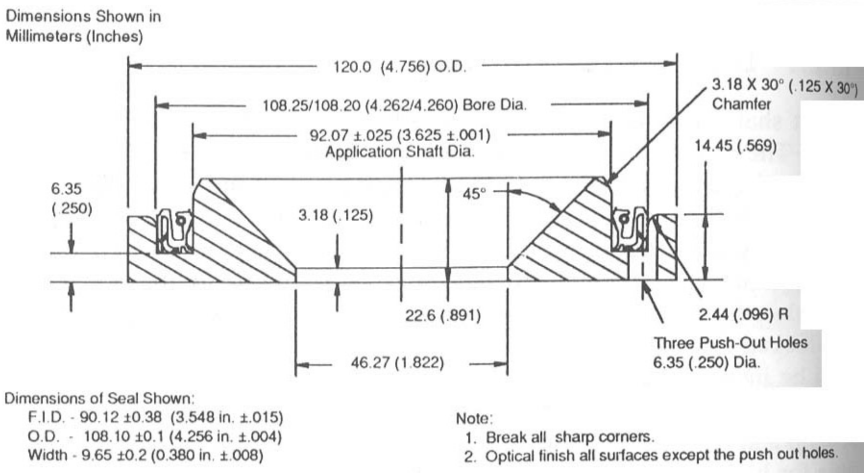

Dimensions Shown in Millimeters (Inches)
120.0 (4.756) O.D.
108.25/108.20 (4.262/4.260) Bore Dia.
92.07 ±.025 (3.625 ±.001) Application Shaft Dia.
3.18 X 30° (.125 X 30°) Chamfer
14.45 (.569)
6.35 (.250)
45°
3.18 (.125)
22.6 (.891)
46.27 (1.822)
2.44 (.096) R
Three Push-Out Holes 6.35 (.250) Dia.
Dimensions of Seal Shown:
F.I.D. - 90.12 ±0.38 (3.548 in. ±.015)
O.D. - 108.10 ±0.1 (4.256 in. ±.004)
Width - 9.65 ±0.2 (0.380 in. ±.008)
Note:
1. Break all sharp corners.
2. Optical finish all surfaces except the push out holes.

Fig. 13.3 Typical self-centering mandrel for a specific seal size. (Courtesy of Rubber Manufacturers Association.)

13.2.2 Transparent Shaft Procedure

The integrity of the lip contact bond can be visually examined through a plastic shaft. The best design is shown in Fig. 13.3 and is self-centering. The shaft is machined and polished to the same diameter as the application shaft. A chamfer or radius is provided to protect the seal lip during installation. The bore diameter of the plastic fixture is equal to or slightly larger than the maximum seal O.D. (0.00-0.05 mm) (0.00-0.002 in.). The bore depth is about half the seal width. If the seal has a rubber O.D., then the depth should encompass at least two beads of the seal O.D. Three holes placed 120° apart are provided so the seal can be pushed out of the fixture after the examination is complete.

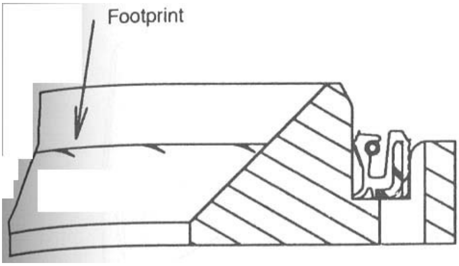


Fig. 13.4 Viewing seal footprint through plastic shaft. The footprint is viewed for acceptable patterns or for discontinuities. (Courtesy of Rubber Manufacturers Association.)

The inspection is carried out by pushing the seal into the bore with the oil side facing up. The contact pattern can be observed on the beveled sides of the inside of the fixture (Fig. 13.4). Any break or discontinuity in the footprint is considered a failure. This procedure is subjective and is unsuitable for high-speed production. It is often used to inspect the “on-shaft” contact patterns of hydrodynamic seals to determine if the patterns are acceptable.

13.2.3 Air Leakdown

The air leakdown measurement has become a useful in-line inspection device for high-speed automated finishing and assembly operations. The seal is placed over a mandrel that is ground to a surface finish of 0.25-0.50 µm (10-20 µin.) Ra (Fig. 13.5). The center of the seal O.D. must be concentric with that of the test mandrel. The seal outer diameter is sealed off, and the cavity formed by the sealing lip and the mandrel is pressurized and sealed. If there are any lip defects, the air will escape and the pressure will drop. For maximum sensitivity, the mandrel should be rotated slowly during the leakdown check.

The amount of air that flows out of the seal can be measured, and the seal is acceptable if the flow rate is less than 50 cm3/min. Pressure decay can also be measured. The seal is acceptable if the pressure drop is less than 1.38 kPa (0.2 psi) within 0.5 sec after the seal is installed on the mandrel and pressure is applied. The applied pressure should be 6.9 kPa (1 psi) below the lower limit of the lip opening pressure specification. Higher pressures could reject good seals by blowing the lip open. An air flow of 50 cm^3/min is equivalent to a pressure drop of 1.38 kPa (0.2 psi) within 0.5 sec if the test cavity volume is 32.8 cm^3 (2.00 $\mathrm{in.}^3$).

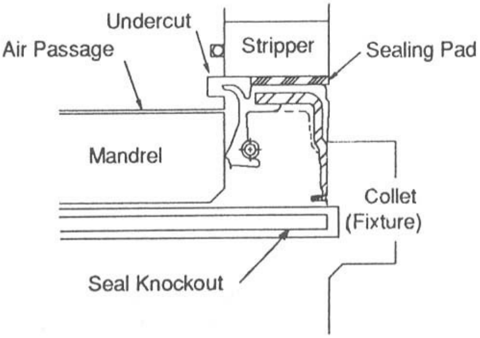


Fig. 13.5 Air leakdown and lip opening pressure test mandrel and fixture. The undercut prevents the auxiliary lip from being curled against the mandrel, which could affect the test results. (Courtesy of Rubber Manufacturers Association.)

13.3 SEAL OUTSIDE DIAMETER

The outside diameter is usually the characteristic of a seal that is most closely toleranced. It is slightly larger than the bore to allow for a press fit. In production quantities it is measured with micrometers, vernier calipers, mechanical dial indicators, electronic indicators, go/no-go ring gauges, laser micrometers, and optical comparators. The RMA referee method of measurement is to use an optical comparator with a magnification of 20x. The seal should be placed flat when using a comparator with the lens axis horizontal to the table (Fig. 13.6a). It should be placed on edge when using a comparator with an overhead lens (Fig. 13.6b). Position the seal so that readings may be taken from two points 180° apart. Seals can have a flare, toe-in, dish, or bulge (Fig. 13.7). Therefore, it is important to mark the exact point where the measurement is taken. Two different people measuring the same seal with optical comparators should obtain readings that are within 0.013 mm (0.0006 in.) for seals with O.D.'s of 135-200 mm (2.5-5.0 in.). The readings should be within 0.025 mm (0.001 in.) for seals with O.D.'s of 270-305 mm (5.0-7.5 in.).

13.4 SEAL SPRUNG AND UNSPRUNG UP INNER DIAMETER

The primary lip diameter is usually the innermost diameter of the seal and is slightly smaller than the shaft diameter. There are several production techniques used to measure the lip. The optical comparator is used as the referee method.

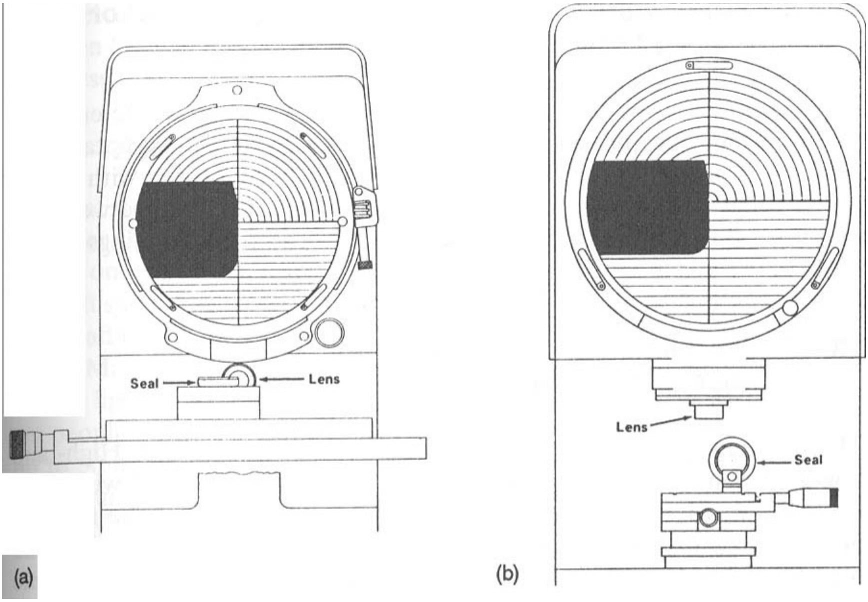


Fig. 13.6 Measuring seal dimensions with an optical comparator. (a) Horizontal lens. Seal placed flat to measure seal O.D.; seal placed on edge to measure lip I.D. (b) Overhead lens. Seal placed on edge to measure seal O.D.; seal placed flat to measure lip I.D. (Courtesy of Rubber Manufacturers Association.)

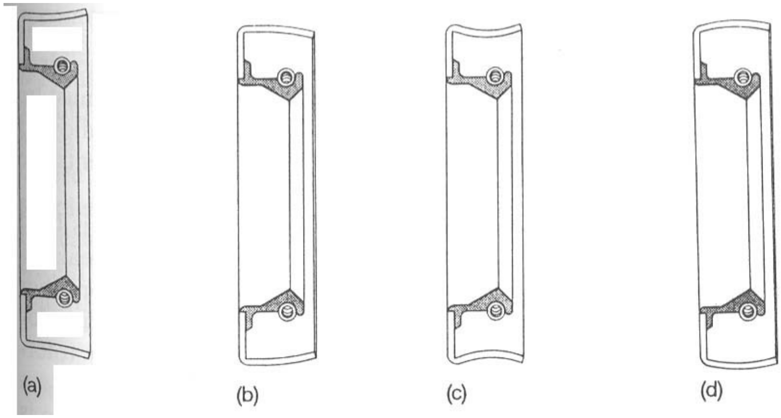


Fig. 13.7 Case deformations. (a) O.D. flare; (b) O.D. toe-in; (c) O.D. dish; (d) O.D. bulge. (Courtesy of Rubber Manufacturers Association.)

13.4.1 Production Method--Light Box with Tapered or Stepped Shaft

This method does not provide an accurate measure of lip diameter in the free state but does provide an approximation that may be acceptable for production checks. A marked shaft with a 1:25 taper is used with a light source below. The seal is lowered until no light can be seen between the seal lip and the shaft. The seal lip diameter is read from markings on the shaft.

13.4.2 Production Method--Light Box with Go/No-Go Gauges

The seal outside diameter is held concentric to a test mandrel. High- and low-limit functional inside diameter mandrels are arranged with light sources underneath them. Test seals are then positioned into the fixtures, noting the light passing between the seal lip and the test mandrel. On the high-limit mandrel the seal lip must preclude all light to be acceptable (indicating that it is smaller than the mandrel), while on the low-limit mandrel the seal lip must allow light to pass. By using mandrels in increments of 0.15 mm (0.005 in.), the actual seal functional lip diameter can be determined.

13.4.3 Production Method--Air Flow

The functional lip diameter of the seal assembly can be measured by means of an air flow between the lip and a mandrel of known size when the air is applied at a standard pressure of 3450 Pa (0.50 psi). Measurements must be taken at a room temperature above 16°C (60°F). The seals must be allowed to stand at room temperature for at least 1 hr before the measurements are made.

The seal is mounted over the mandrel in a retainer fixture and is held concentric to the mandrel within 0.05 mm (0.002 in.) TIR. Air leakage around the seal case and the mandrel pilot is prevented by O-rings or other suitable gaskets. The test mandrel diameters are the same as specified for the maximum and minimum lip diameters. The mandrel diameter tolerances are ±0.013 mm (0.0005 in.), and the surface roughness is 0.38 μm (15 μin.) or less. The seal is placed in the test fixture so the air pressure is applied to the outside face and the air flow between lip and mandrel is measured. The air flow between the seal lip and the minimum diameter test mandrel must be equal to or greater than 10,000 cm^3/min. The air flow between the seal lip and the maximum diameter test mandrel must be equal to or less than 10,000 cm^3/min.

13.4.4 Referee Method--Optical Comparator

The seal is placed in the fixture of a 20x optical comparator so that the lip diameter is at a right angle to the comparator lens. The seal should be placed on edge when using a comparator with the lens axis parallel to the table. It should be placed flat on the table when using a comparator with an overhead lens. The seal is positioned to take the reading at two points 180° apart. Mark the exact place where the measurement is taken because all sealing lips are out of round to some degree. Measurements by two different people using the same seal should be within 0.025 mm for seal lip I.D.'s up to 50.8 mm (2.000 in.); 0.050 mm for seal lip I.D.'s between 50.8 and 101.6 mm (2.000-4.000 in), and 0.076 mm (0.003 in.) for seal lip I.D.'s between 101.6 and 152.4 mm (4.000-6.000 in.).

13.5 SEAL WIDTH

The seal width is measured with micrometers, vernier calipers, bridge gauges, and indicating gauges. An advantage of indicating gauges is that the seal is supported by a relatively fiat base and excessive out-of-flat or rock can be observed. A mechanical dial gauge accurate to 0.013 mm (0.0005 in.) with a base large enough to support the entire seal is the recommended referee measuring device.

The seal is placed on the base with the indicator point directly over the measurement point. The exact place where the measurement is made should be marked because the width may vary around the circumference. If a seal with a toe gasket is being measured, some deflection may occur where the indicator foot rests on the gasket. In this case, the user and supplier should agree upon the amount of weight that will be applied by the indicator. Measurements by the two parties on the same seal should be within 0.025 mm (0.001 in.) for widths up to 12.7 mm (0.500 in.) and 0.051 mm (0.002 in.) for seal widths greater than 12.7 mm (0.500 in.).

13.6 RADIAL WALL VARIATION

The radial wall dimension (RWD) is the distance from the seal O.D. to the lip I.D. The radial wall variation (RWV) is the difference between

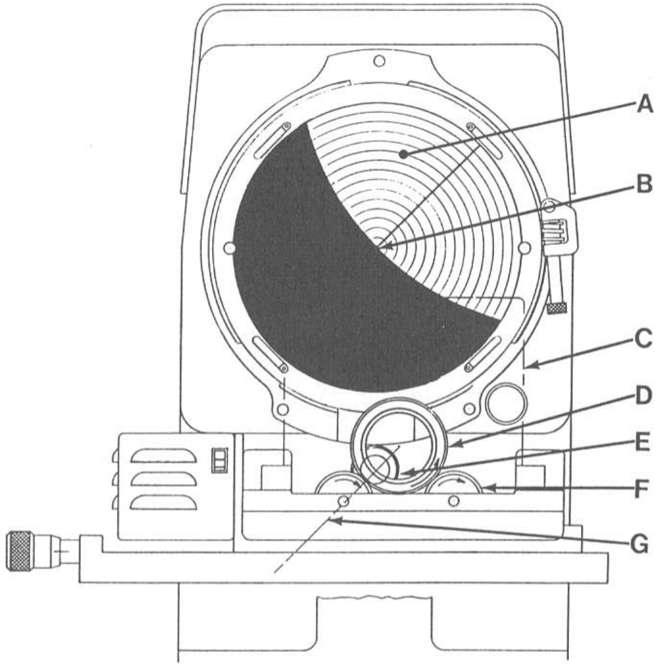


Fig. 13.8 RWV fixture and optical comparator. (A) Optical comparator screen equipped with radius and/or grid chart or precision overlay. (B) Point at which measurement is taken through 360° rotation. (C) Glass plate support. (D) Seal. (E) Lens. (F) Supporting rollers. (G) Imaginary line showing seal alignment with lens and roller. (Courtesy of Rubber Manufacturers Association.)

the largest and smallest radial wall dimensions of a seal. The acceptable measuring technique for both production quantities and the referee method is an RWV fixture used in conjunction with a 20x optical comparator (Fig. 13.8).

The RWV fixture is mounted on the optical comparator table, and the seal is placed on the rollers using the glass plate for support. The seal is slowly rotated and the comparator is adjusted so the highest point of the projected image is at zero on the comparator grid screen. The seal is again rotated. The largest difference from the zero point is the radial wall variation. The accuracy of the method when measured by two different people with the same seal is within 0.025 mm (0.001 in.) for seals up to 76.2 mm

(3.000 in.) and 0.050 mm (0.002 in.) for seals between 76.2 and 152.4 mm (3.000 and 6.000 in.).

13.7 SECONDARY LIP DIAMETER

The secondary lip diameter is usually larger than the primary lip diameter and cannot be measured with an optical comparator unless the primary lip is removed. This is not recommended, because stresses may be relieved and the secondary lip diameter may change. Go/no-go gauges, tapered cones, or light box gauges are sometimes used. The recommended method uses a 20x toolmaker's traversing microscope. The seal is placed on the table with the secondary lip facing the lens. The seal is positioned so the cursor will travel through the seal center and indicate the distance between two points on the secondary lip I.D. that are exactly 180° apart. The seal should be marked so the exact points of measurement are known. The instrument is zeroed at one point on the secondary lip. The table is traversed to the point on the secondary lip that is opposite the starting point. The distance traveled is the secondary lip diameter.

The measurements made by two parties on the same seal should be within 0.025 mm (0.001 in.) for secondary lips up to 50.8 mm (2.000 in.), 0.051 mm (0.002 in.) for secondary lips between 50.8 and 101.6 mm (2.000 and 4.000 in.), and 0.076 mm (0.003 in.) for secondary lips between 101.6 and 152.4 mm (4.000 and 6.000 in.).

13.8 SPRING AXIAL POSITION

The spring axial position or R value is the axial distance between the projected intersection of the inside and outside lip surfaces and the centerline of the spring coil diameter (center plane of the spring) with the spring in position and the seal located on the shaft. This distance is a constant value for some seal designs but varies with designs that have wavy contact patterns. When measuring the R value of seals of the latter design, it is recommended that two measurements be made, one at the high end of the wave and one at the low end. There is no quick production method available to measure R value.

The seal to be measured is placed in a fixture that simulates the shaft and housing bore assembly [212], This fixture can be constructed of various materials suitable for polishing and adhesion to a potting material. The shaft is held concentric with the housing bore within 0.05 mm (0.002 in.) TIR. The seal and fixture are then encapsulated in a potting material such as a dental casting plaster or an epoxy having a shrinkage of not more than 2%. The assembly is then cross-sectioned through its center in a vertical plane and polished. Care must be taken not to distort the seal during the sectioning operation. The spring axial position is now measured by viewing the cross section using either a reflective optical comparator or a tool-maker's microscope.

13.9 CONTACT LINE HEIGHT VARIATION

Contact line height variation is the difference between the maximum and minimum axial dimensions from the seal contact line to the outside face. Seal designs with wavy lips have a variation in contact heights that is equal to the wave amplitude. The recommended method of measuring contact line height variation is to use a microscope and prism (Fig. 13.9). In this method, the contact line height variation is read directly as the seal is rotated above a fixed center. The method is easy to use and offers a good degree of repeatability. The equipment is inexpensive and does not require tooling for each seal size. Since the seal contact line is measured in its free position, handling should be minimized to avoid distortion. One method of minimizing distortion is to insert a shaft-size mandrel concentric with the seal O.D. prior to measuring.

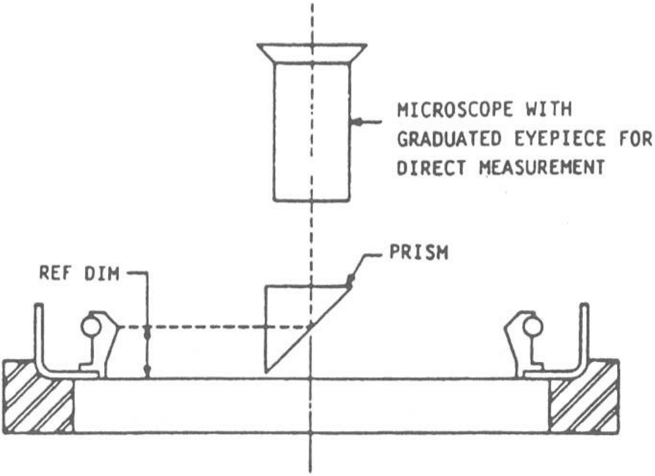


Fig. 13.9 Measurement of contact line variation.

13.10 SEALING EDGE ROUGHNESS

Sealing *edge roughness* is the condition of the surfaces on the seal that form the seal shaft interface. The roughness may be caused by voids, poor trim, dirty mold surfaces, foreign material in the compound, etc. The Society of Automotive Engineers [195] differentiates between contact height variation and contact height roughness by stating, "Any deviation existing in a small increment of seal contact line length (approximately 6 mm (0.236 in.) is sealing edge roughness." Production methods of measuring sealing edge roughness vary from optical examination with magnification to mechanical air leakdown devices. There is no recognized referee method.

13.11 LIP OPENING PRESSURE

Lip opening pressure (LOP) is used to measure the consistency of production parts. It is sometimes used in conjunction with the leakdown procedure to determine lip integrity (Fig. 13.10). The specification for LOP is usually determined by statistically averaging readings from samples of the first three production runs of a new production seal. LOP is defined to be the pressure applied to the air side of the seal that will lift the lip and allow 10,000 cm^3/min of air to flow past the lip. A test mandrel with a surface finish of 0.38 µm (15 µin.) Ra or less is prepared that is similar to the device shown in Fig. 13-5. The seal is installed in the fixture, and pressure is increased from zero to its maximum value within 3 sec. The pressure on the air side will lift the lip and allow air to pass. When 10,000 cm^3/min is flowing by the lip, the pressure is recorded. When both LOP and seal lip I.D. measurements are made consecutively, the seal lip I.D. must be measured first to avoid errors due to deformation of the material. Repeat measurements on the same seal must not be taken within 16 hr of a lip opening pressure measurement.

13.12 RADIAL LOAD

The radial load of an oil seal is defined as the load that an oil seal exerts on the shaft and is the sum of the central acting forces around the seal lip. The radial load is generally composed of two principal elements: the spring and the sealing element [17]. The sealing element load is created by two distinct modes of stress within the material: the hoop tension created by stretching of the lip and the beam load created by radial deflection outward of the seal lip element (Fig. 13.11).

Simple equations can be used to obtain an estimate of radial load (RL) and lip opening pressure (LOP). Some of the terms in the equations are defined in Fig. 13.12.

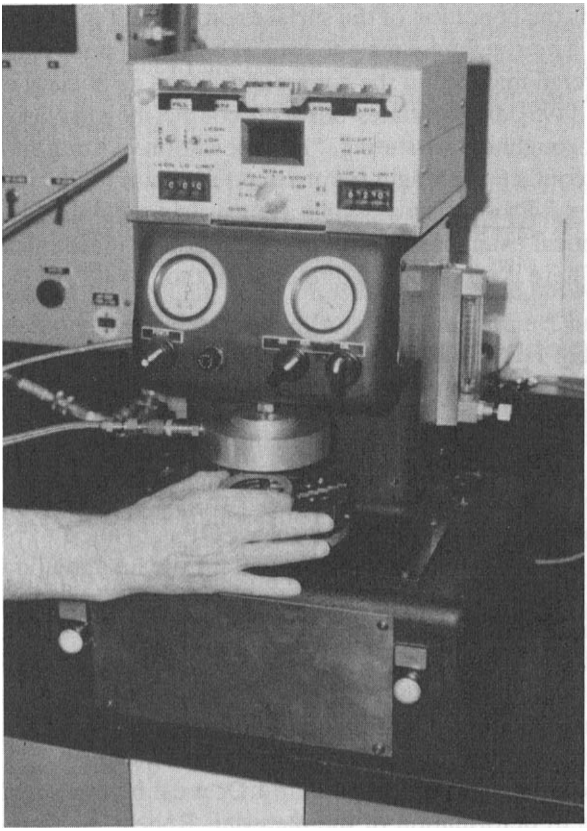


Fig. 13.10 Machine to measure lip opening pressure (LOP) and leak down rate.

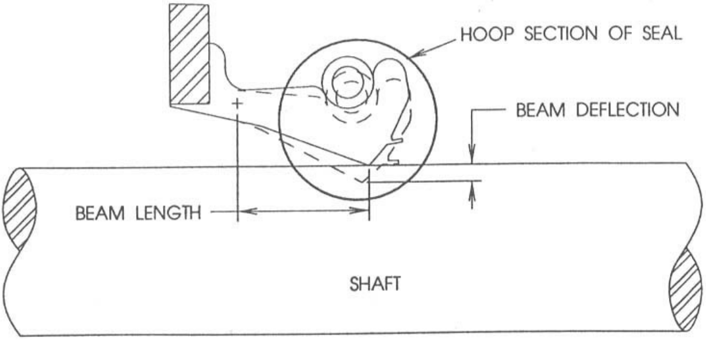

HOOP SECTION OF SEAL
BEAM DEFLECTION
BEAM LENGTH
SHAFT

Fig. 13.11 Seal radial force is generated by the spring and the elastomeric beam and hoop forces.

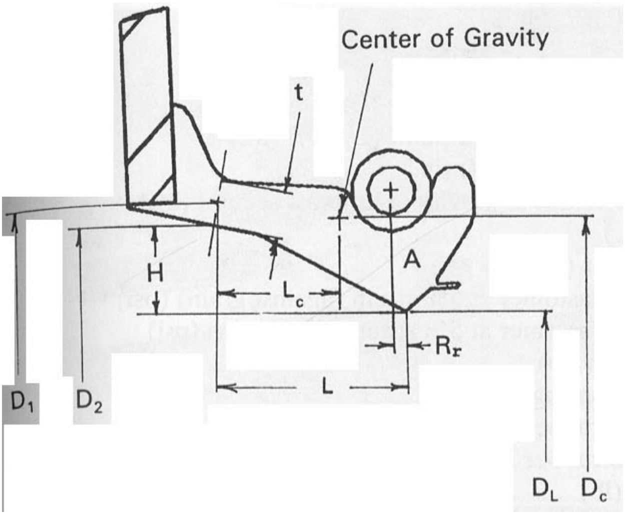


Fig. 13.12 Geometrical definition of terms to define radial load (RL) and lip opening pressure (LOP).

$$F_S = 2\pi \left[1 - \frac{R_r}{L} \right] F \quad \text{(lb)} \tag{13.5}$$

$$F_B = \frac{\pi}{8} E_W D_1 (D_S - D_L) \left(\frac{t}{L} \right)^3 C \quad \text{(lb)} \tag{13.6}$$

$$F_R = 2\pi E_A A \left(\frac{D_S - D_L}{D_C} \right) \left(\frac{L_C}{L} \right)^2 C \quad \text{(lb)} \tag{13.7}$$

$$F_T = F_S + F_B + F_R \quad \text{(lb)} \tag{13.8}$$

$$R_L = \frac{F_T}{\pi D_S} = \frac{2F}{D_S} \left(1 - \frac{R_r}{L} \right) + C \left(1 - \frac{D_L}{D_S} \right) \left[\frac{E_W D_1}{8} \left(\frac{t}{L} \right)^3 + \frac{2E_A A}{D_C} \left(\frac{L_C}{L} \right)^2 \right] \quad \text{(lb/in.)} \tag{13.9}$$

$$\text{LOP} = \frac{2\,F_T}{\pi L D_S \left\{ \left(1 + \frac{2H}{D_S} \right) + \left(\frac{H}{L} \right)^2 \left(1 + \frac{H}{D_S} \right) \right\}} \quad \text{(psi)} \tag{13.10}$$

where

A = cross-sectional area of lip (in.^{2})
C = elastomer compression factor (about 0.85)
D1 = diameter to center of flex point (D_2 + t) (in.)
D2 = heel diameter (Ds + 2H) (in.)
DC = diameter to center of gravity of lip cross section (in.)
DL = lip I.D. (in.)
DS = shaft diameter (in.)
EA = modulus of elastomer at 5% strain (against grain) (psi)
EW = modulus of elastomer at 5% strain (with grain) (psi)
F = spring tension (lb)
FS = force due to spring (lb)
FB = force due to rubber beam deflection (lb)
FR = force due to rubber hoop forces (lb)
FT = total force (lb)
H = distance from contact point to heel (in.)
LC = distance from flex point to center of gravity (in.)
L = distance from flex point to lip contact point (in.)
Rr = R value (in.)
RL = radial load (lb/inch of shaft circumference)
t = flex or beam thickness (in.)

Note:
1 newton (1 N) = 0.017355 lb
1 N/cm = 0.0068125 lb/in.
1 N/cm^2 = 0.00269 lb/in.^{2} (psi)

Later methods [68] used numerical techniques to predict load. Modern techniques use finite-element analysis to obtain the force and pressure distributions [40, 205].

Radial load is typically measured with a split-shaft device (Fig. 13.13) [155, 200]. The principle is illustrated in Fig. 13.14.

$$F_i = 2\int_0^{\pi/2} F\,R_S \sin\alpha\, d\alpha = FR_S \tag{13.11}$$

$$R_L = \frac{F_i}{2R_S} = \frac{F_i}{D_S} \tag{13.12}$$

$$T = 2\pi R_S R_L \tag{13.13}$$

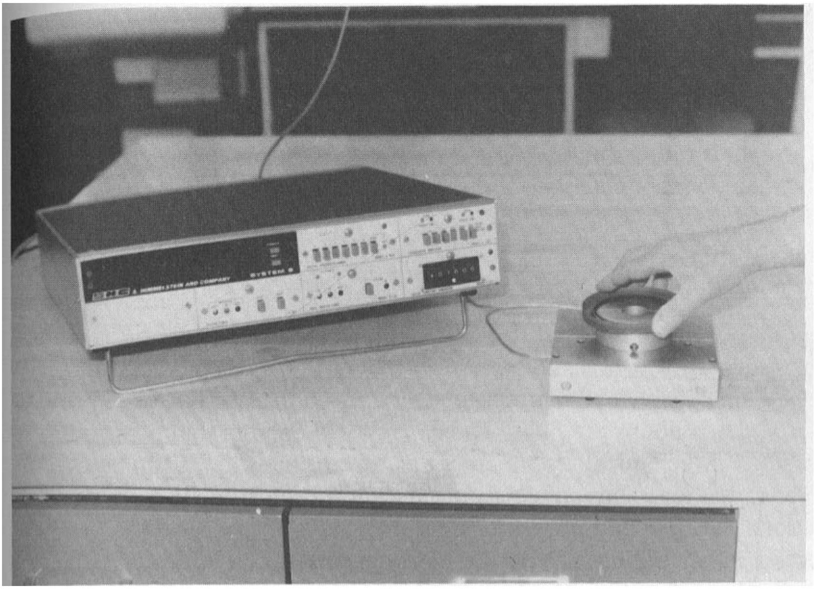


Fig. 13.13 Electronic radial load measuring system. (Courtesy of Chicago Rawhide.)

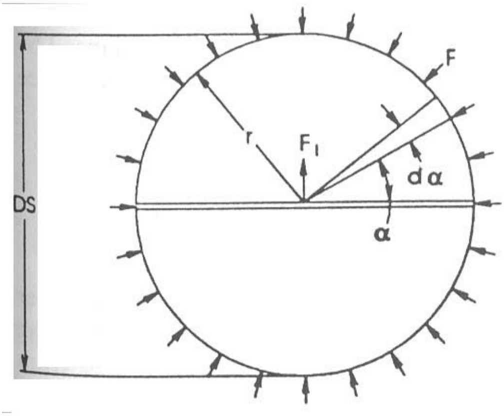

F
r
F_1
$d\alpha$
DS
α

Fig. 13.14 Split shaft principle.

where

D_s = shaft diameter (in.)
F = incremental force exerted by seal lip (lb)
Fi = force sensed by instrument (lb)
R_s = shaft radius (in.)
α = angle from center of shaft split to any point on shaft surface (degrees)
$d\alpha$ = differential angle (degrees)
T = total radial force of seal (lb)

One half of the unit is rigidly attached to a table; the other half is mobile and is attached to a measuring system that will detect and record force. There is a small gap (0.38 mm; 0.015 in.) between the two halves to prevent contact and erroneous readings. The gap is small enough that the reading is not affected. Mechanical, pneumatic, and electronic systems can be used to measure radial load, but an electronic system is preferred. Very small shaft deflections can provide accurate measurements. Some machines can be immersed in an oil bath (Fig. 13.15), and the radial load can thus be determined as a function of oil type, oil immersion time, and temperature.

The seal is placed over a split mandrel that is the same diameter and surface finish as the application shaft. The radial load will jump to a high value and then decay as the elastomeric lip undergoes stress relaxation (Fig. 13.16). It is recommended that the seal remain on the machine for 30-60 sec until the reading stabilizes. A radial lip seal exerts an incremental lip force. The force components perpendicular to the split are balanced by the force FL, which the instrument actually measures in appropriate force units per unit of shaft diameter.

13.13 SEAL O.D. COATING MEASUREMENTS

13.13.1 Types of Seal Coatings

Thin coatings of resin-based materials are sometimes placed on the outer diameter of radial lip oil seals with metallic cases to provide sealing if the application bore is rough. The thickness of the coating can range from 0.005 to 0.076 mm (0.0002-0.0003 in.). The metallic O.D. is often treated before the bore sealant is applied. The metal surface is blasted or phosphated, and a thin layer of rubber bonding adhesive is applied. These subcoats have a small but significant thickness and must be considered when measuring bore sealant thickness. The thickness of the subcoats may range from 0.003 to 0.018 mm (0.0001-0.0007 in.) and will depend upon the type of phosphate (iron or zinc) used and the consistency of the cement.

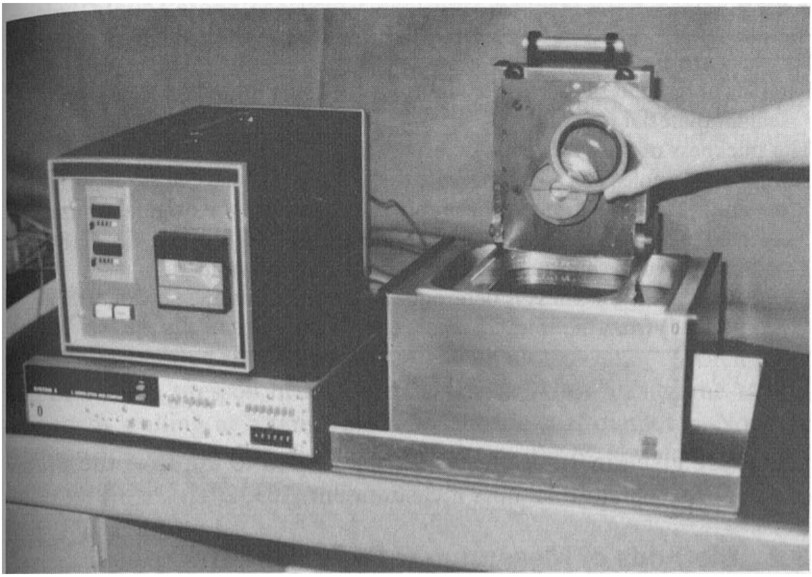


Fig. 13.15 Measuring radial load in hot oil. (Courtesy of Chicago Rawhide.)

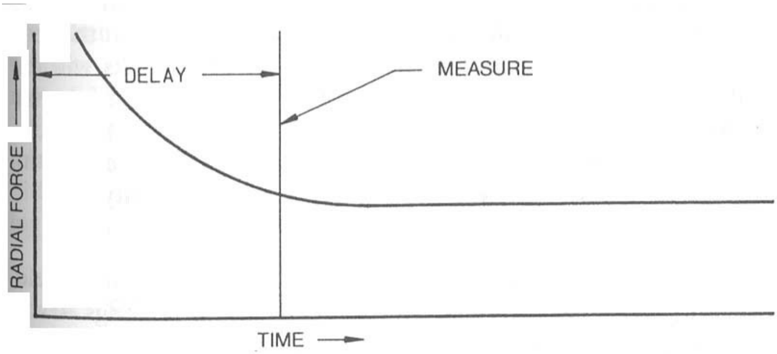


Fig. 13.16 Stress relaxation curve for a typical elastomer.

Table 13.3 Procedure to Estimate Thickness of Subcoatings on Metal Cases

1.	Subcoats are applied to a given lot of metals.
2.	Samples of molded seals are selected from the given lot before the O.D. sealant material is applied.
3.	The thickness of the subcoats is determined.
4.	Additional samples are obtained from the given lot of seals after the O.D. sealant is applied. It may be desirable to tag and use the original samples selected in step 2.
5.	The thickness of the subcoats and O.D. sealant is determined.
6.	The O.D. sealant thickness is determined by subtracting the subcoat thickness (step 3) from the total thickness (step 5).

Source: Ref. 163.

The procedure given in Table 13.3 is recommended to improve the precision of the O.D. sealant thickness measurement [163, 201].

13.13.2 Methods of Measuring O.D. Sealant Thickness

Electronic

The electronic method for measuring O.D. sealant thickness will only work with steel cases. The coating is exposed to a low-frequency magnetic-inductive probe, and three equally spaced readings are taken about the seal outside diameter. These readings are then averaged. The results will include the thickness of the O.D. coating and subcoats. If desired, the subcoat thickness can be determined and subtracted from the total thickness. The electronic method of measuring thickness is preferred over others because it is easy to perform and provides good reproducibility.

Magnetic

The magnetic method will work only with steel cases. Magnetic hand-held gauges (i.e., Biddle gauges) are available to obtain quick readings. Care must be exercised because the readings are dependent upon operator skill. The gauge must be held perpendicular to the surface. Three equally spaced readings are taken about the seal outer diameter and then averaged to give the total thickness of O.D. sealant and subcoats. If desired, the subcoat thickness can be determined and subtracted from the total thickness.

Mechanical

The mechanical method is recommended for use with nonmetallic seal cases. It can also be used with steel cases. The O.D. of the seal is carefully marked and measured in three places, using the method described in Ref. 163. The O.D. coating is removed by soaking the seal in a solvent such as methyl ethyl ketone (MEK). Care must be exercised not to abrade or damage the seal outer diameter. After the O.D. sealant is removed, the seal O.D. measurements are repeated exactly as before. The average difference between the two sets of readings is determined and divided by 2 to give the O.D. sealant thickness. This method is lengthy, and soaking may remove part but not all of the subcoatings.

13.13.3 Method of Measuring Bond Strength

X Test

An X of approximately 0.25-0.5 in. (6.35-12.7 mm) in height is cut through the coated surface to the metal with a razor blade. Scotch tape (3M 600 or 610) is pressed firmly onto this area. After 5 min, the tape is removed by pulling at a tangent to the outside seal diameter. None of the coating other than that disturbed by cutting the X should adhere to the tape, no matter how fast the tape is pulled.

Cross-Hatch Test

A segment of the coated O.D. is cross-hatched by cutting through the coated surface to the metal with a razor blade. Sufficient cuts are made in each direction to result in approximately 100-150 small squares of about 0.040 in. (1.00 mm) on a side. Scotch tape (3M 600 or 610) is applied to the cross-hatched surface. After 5 min, the tape is removed slowly, and the number of sections with intact O.D. coatings is determined. If 80% or more of the sections are intact, the bond strength is acceptable.

The cross-hatching operation must be performed with care to avoid loosening the O.D. coating material. If it does loosen, then it will be difficult to differentiate between poor adhesion and material loosened during cutting. The test will be invalid and should be repeated.

13.14 SEAL CROSS SECTION MEASUREMENT

Maintaining radial lip seal geometry is important to seal function. Six dimensions have been defined that when properly measured will ensure the conformation of production seal cross sections to the original print dimensions (Fig. 13.17) [151]. Deviations from the print dimensions may mean that the mold is not registering properly, the mold is not fully closed, or lip trimming procedures are not within process specifications. The included angle α will not vary if the lip is molded and the mold is cut to proper specifications. Variations in α may occur if the seal is trimmed. The distance from the outside lip surface to the spring groove radius (B) will vary if the mold is not in proper register or if it is not closing properly. The

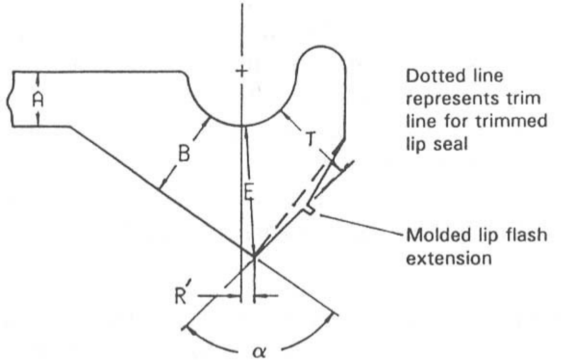

A
B
T
E
R'
α
Dotted line represents trim line for trimmed lip seal
Molded lip flash extension

Fig. 13.17 Definition of lip seal cross-sectional variables, α = the maximum allowable included angle, B = distance from the outside lip surface to the spring groove radius; E = distance from the contact point to the spring groove radius, T = the minimum distance from the trimmed or molded surface on the spring retaining lip to the spring groove radius; A = the flex section thickness; R' = the relationship of the contact point to the centerline of the spring groove radius when the seal lip is in a free position. (Courtesy of Rubber Manufacturers Association.)

distance from the contact point to the bottom of the spring groove (E) and the minimum distance from the spring groove radius to the trimmed or molded surface on the oil side of the seal (T) will vary if mold closure and/ or register and/or trim procedures are not properly maintained. Variations in mold register will cause variations in the flex section thickness A. The distance from the contact point to the spring groove center in the free position (R') will vary with variations in mold closure and trimming process procedures.

The measurement procedure begins with the preparation of a cross section from a production seal sample. The section must be thin to reduce any distortion due to the lip diametrical curvature. Reflective magnifying methods coupled with a thin section eliminate this problem. Three templates from a typical 10x drawing of the given seal cross section are transposed onto a transparent film (Fig. 13.18). The solid line of Fig. 13.18a is aligned

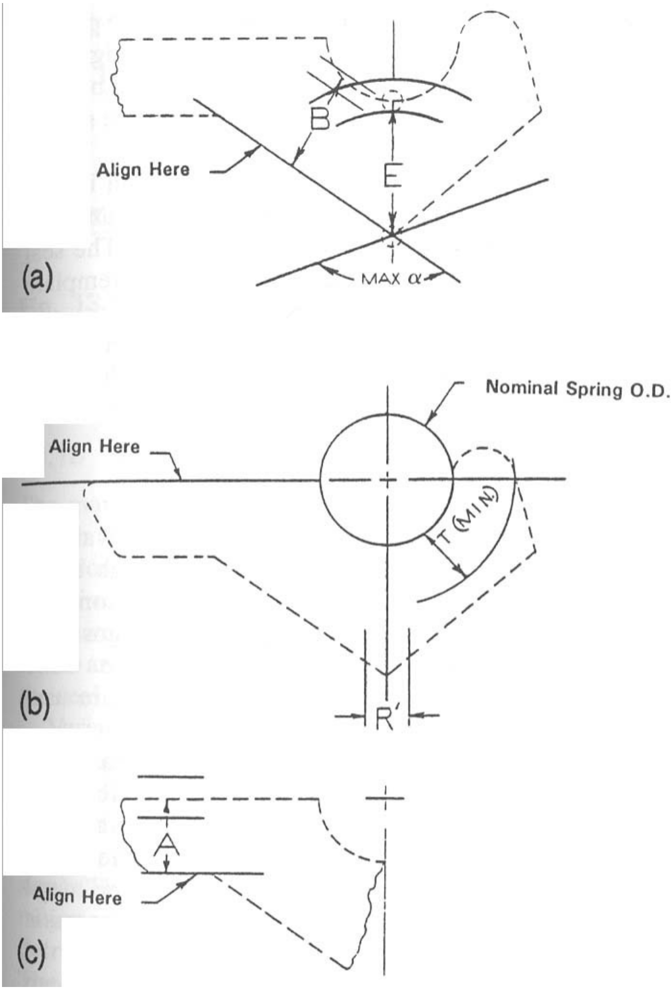

(a)
B
E
Align Here
MAX α
(b)
Nominal Spring O.D.
Align Here
T (MIN)
R′
(c)
A
Align Here

Fig. 13.18 Measuring α BETAR′. (Courtesy of Rubber Manufacturers Association.)

with the barrel surface of the production seal cross section using the projection on a comparator screen. The contact point is positioned to align with the contact point of the template. The parameters α, B, and E can be read directly from the comparator, and any deviations from the template should be noted.

The seal cross section is next aligned over the template shown in Fig 13.18b and is adjusted to align with the solid line. The template and seal cross section are aligned, and values of T and R are measured. The seal cross section is then positioned to align with the solid line of the template (Fig. 13.18c), and the flex section thickness A is read directly.

Seal designs with wavy lips may result in cross-sectional variations around the seal lip circumference, α BETAR' measurements must be made at a reference point that is agreed upon by seal users and suppliers.

13.15 BOND TESTS

Bond failure between the rubber seal element and the case will always result in catastrophic seal failure. The seal will experience high stress levels at the bond area during installation. High-pressure applications can also exert high stress in this region, and most high-pressure designs have a plastic backup ring to take this load and prevent bond rupture. Once the seal is installed in a low-pressure environment, stress levels are usually low.

Testing seals to determine bond strength is very subjective. Pliers are used to grip the elastomeric lip, and the lip is pulled in the axial direction until separation occurs. If the rubber ruptures instead of being pulled away from the metal, then the bond is good. If the rubber separates from the metal and there is rubber residue on the metal, then the bond is good. If the metal surface is clean without a rubber residue, then the bond is bad. The process is difficult to control, subjective, and highly operator-dependent.

A more controllable method of determining bond strength has been developed [162, 199]. A clamping device is attached to a standard tension test device. The jaws of the clamping device are attached to the seal lip (Fig. 13.19), and the cross head is raised at a rate of 500 mm/ min (20 in./min) until failure occurs. The jaws of the mechanical clamp should be adjusted so they will lock with a force of 4.4-8.8 N (1-2 lb). The force required to induce the failure is recorded. A statistical sample of a new production seal can be obtained and tested to establish the minimum acceptable load level for failure. Quality engineers can use this load level as a criterion for bond failure when checking future production lots.

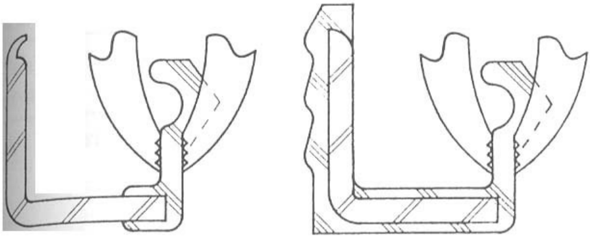


Fig. 13.19 Method of clamping seal lip with mechanism to determine bond strength. (Courtesy of Rubber Manufacturers Association.)

13.16 VISUAL VARIATION

The manufacture and handling of shaft seals may generate minor damage known as visual variations at various locations on a seal. Most of the visual variations will not result in functional failure. The significance of visual variations is dependent upon the primary function of the seal, the location of the variations, and the magnitude of the variation. The location of seal zones and typical visual variations are shown in Fig. 13.20. Definitions of the seal zones are given in Table 13.4 [195].

Variation magnitude is determined by visual inspection with magnification. A guide to visual variations and their magnitudes has been published by the Rubber Manufacturers Association [157]. A guide to the significance of the variations appears in Table 13.5, which is divided into three sections by primary function (oil retention, grease retention, dirt exclusion). The guide can be used to determine if the visual variation is critical, major, or minor. The designation critical signifies that the variation is detrimental to fluid or grease retention and/or contaminant exclusion. The use of seals with critical variations is not recommended. Major variations may or may not be detrimental to seal function. The use of seals with major variations must be determined by a careful review of service requirements, the magnitude of the visual variation, and the consequences of seal failure.

All major variations should be reviewed prior to initiation of inspection standards. Those areas designated major that are detrimental to the specific application under consideration should be redesignated as critical. The decision as to the detrimental nature of each variation should be predetermined by testing. Minor variations are not detrimental to seal function, and the use of seals with minor variations is permissible.

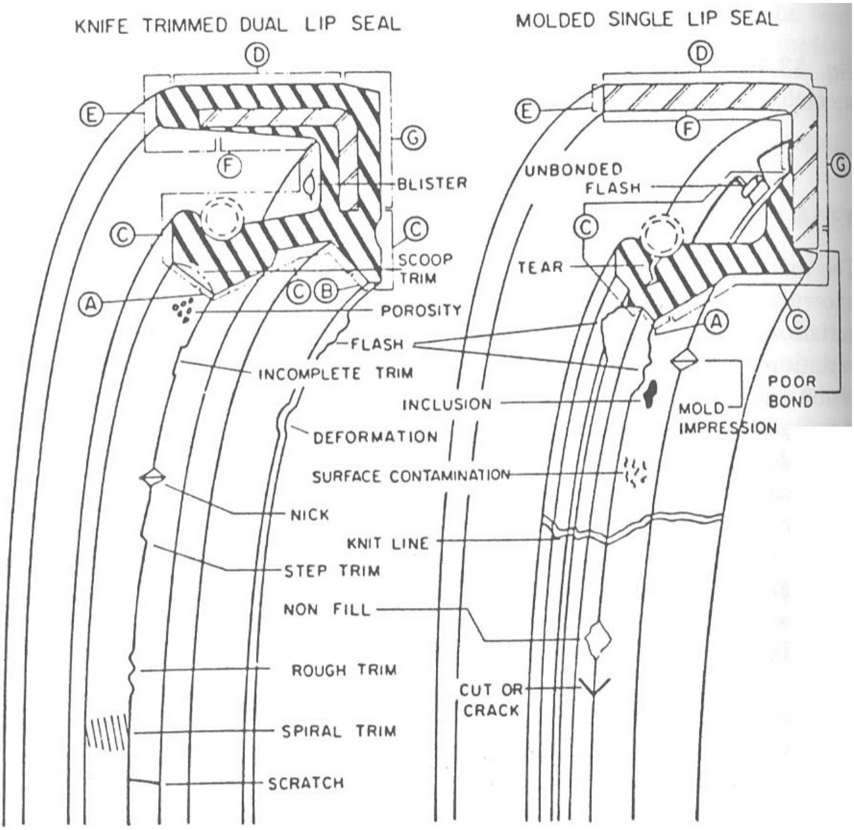

KNIFE TRIMMED DUAL LIP SEAL
MOLDED SINGLE LIP SEAL
D
E
F
G
BLISTER
C
SCOOP TRIM
A
C
B
POROSITY
FLASH
INCOMPLETE TRIM
DEFORMATION
NICK
STEP TRIM
ROUGH TRIM
SPIRAL TRIM
SCRATCH
D
E
F
G
UNBONDED FLASH
C
TEAR
A
C
POOR BOND
MOLD IMPRESSION
INCLUSION
SURFACE CONTAMINATION
KNIT LINE
NON FILL
CUT OR CRACK

Fig. 13.20 Typical visual variations. (Courtesy of SAE.)

Table 13.4 Description of Seal Area

Seal area	Area name	Description
A	Primary lip contact area	The lip of any single-lip seal is the primary lip. If a multiple-lip seal is used for fluid retention, the fluid-retaining lip is the primary lip and the contaminant-excluding lips are secondary lips. If a multilip seal is not used to retain a fluid, the primary lip would be the most critical lip. A seal could have two primary lips if two fluids are separated by a multilip seal. The maximum wear anticipated on the primary lip establishes the boundaries of area A.
B	Secondary lip contact area	The secondary lip or lips are intended for dirt, dust, or other contaminant exclusion. The maximum wear anticipated on the secondary lip or lips establishes the boundaries of area B.
C	Rex section	The flex section, the remainder of the primary and secondary lips not included in areas A and B, and any other flexing and critical areas constitute area C.
D	Seal O.D.	The seal O.D. where contact is provided with the mating bore constitutes area D.
E	Seal end face	The end face of the seal that serves to locate the seal axially in the mating bore constitutes area E. On some applications, areas E and G could be reversed from those shown in Fig. 13.18.
F	Interior areas	All nonfunctional interior areas constitute area F.
G	Exterior areas	All nonfunctional exterior areas constitute area G. On some applications, areas E and G could be reversed from those shown in Fig. 13.18.

Source: Ref. 195.

Table 13.5 A Guide to Visual Variations

Type of application variation	Primary function								
	Oil retention			Crease retention			Dirt exclusion		
	Critical	Major	Minor	Critical	Major	Minor	Critical	Major	Minor
Blister	A	B,C,D,E	F,G	A	B,C,D	E,F,G	--	A,B	C,D,E,F,G
Porosity	A	B,C	D,E,F,G	A	B,C	D,E,F,G	--	A,B	C,D,E,F,G
Nonfill	A	B,C,D,E	F,G	A	B,C,D,E	F,G	--	A,B,C,D	E,F,G
Mold impression	A	B,D	C,E,F,G	A	B	C,D,E,F,G	--	A,B	C,D,E,F,G
Knit line	--	A,B,C,D,E	F,G	--	A,B,C,D	E,F,G	--	A,B,C	D,E,F,G
nclusion	A	B,C,D	E,F,G	A	B,C,D	E,F,G	--	A,B	C,D,E,F,G
Surface contamination	A	B,C,D,E,F	G	A	B,C,D,E,F	G	--	A,B,C,E,F	D,G
Tear	A	B,C,D,E	F,G	A	B,C,D,E	F,G	--	A,B,C,D,E	F,G
Nick	A	B,C,D,E	F,G	A	B,C,D,E	F,G	--	A,B	C,D,E,F,G
Scratch	--	A,B,D	C,E,F,G	--	A,B,D	C,E,F,G	--	A,B	C,D,E,F,G
Deformation	--	A,B,C,D,E	F,G	--	A,B,C,D,E	F,G	--	ATB,C,D	E,F,G
Cut or crack	A	B,C,D,E	F,G	A	B,C,D,E	F,G	--	A,B,C	D,E,F,G
Flash	--	A,B,C,D,E	F,G	--	A,B,C,D,E	F,G	--	A	B,C,D,E,F,G
Unbonded flash	--	A,B,C,D,E,F	G	--	A,B,C,D,E,F	G	--	A,B,C,D,E,F	G
Poor bond	C	D,E	F,G	C	D,E	F,G	C	D,E	F,G
Scoop trim	--	A,B	--	--	A,B	--	--	--	A,B
Incomplete trim	--	A,B	--	--	A,B	--	--	A	B
Spiral trim	--	A,B	--	--	A,B	--	--	--	A,B
Rough trim	--	A,B	--	--	A,B	--	--	A	B
Step trim	--	A,B	--	--	A,B	--	--	A	B

Source: Ref. 195.

13.17 GARTER SPRING MEASUREMENTS

Key garter spring parameters are the assembled inner diameter (AID), the load at a predetermined extension, and the shape of the load curve [154]. The nib joint strength, the gap, and the tendency for spring wind-up should also be checked. The assembled inner diameter is inspected with the taper gauge shown in Fig. 13.21. The spring must fit on the taper that is defined by the minimum and maximum values of the assembled spring inner diameter. The device is also used to test the strength of the nib joint. The spring must expand over a diameter that is 1.35 times greater than the nominal AID without separation. The maximum allowable gap at the nib joint is two wire diameters.

The spring may have a tendency to deform from a flat position into a figure 8. This results from wind-up during assembly and can be controlled or eliminated by back-winding the spring ends before assembly. Springs that form a figure 8 are acceptable if they snap back to their original diameter when dropped approximately 300 mm (12 in.) onto a flat hard surface.

The load curve is obtained with an extension device that records the force required to elongate the spring. The garter spring is cut at a point approximately 180° away from the nib joint, and the ends are installed in the extension device. The nib integrity is tested while the load curve is being generated.

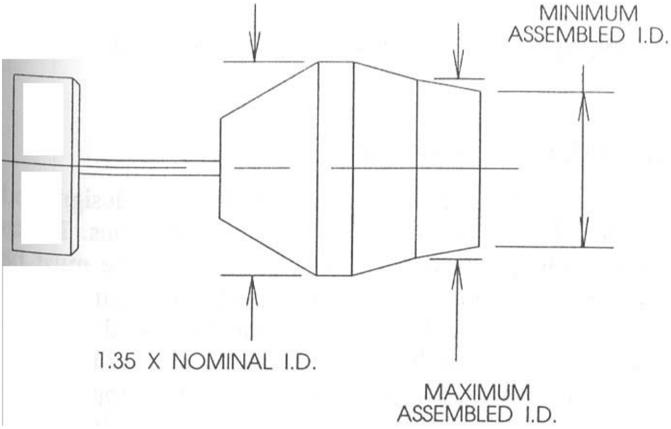


Fig. 13.21 Spring inspection gauge. (Courtesy of Rubber Manufacturers Association.)

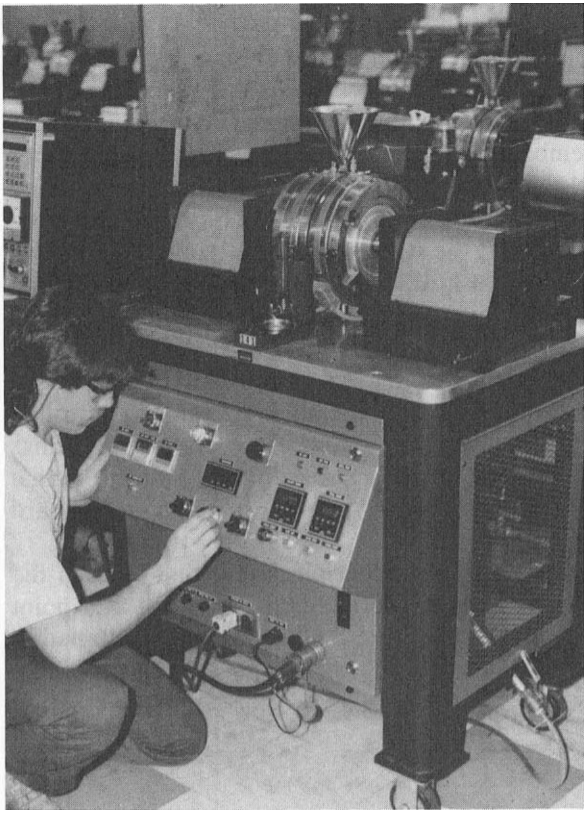


Fig. 13.22 Bench test machine for seal audit tests. (Courtesy of Chicago Rawhide.)

13.18 FUNCTIONAL BENCH TESTS

Functional bench tests are used to evaluate new radial lip seal designs and materials before being placed into service in actual applications. Bench tests are also used to audit production parts (Fig. 13.22). Care must be taken to ensure that the test equipment meets the specifications and industry standards described in Section 7.5.1. The test conditions used to evaluate the seals should not exceed the limits of the seal design or material. Recommended test parameters for evaluating seals for various applications appear in Tables 13.6-13.8. The following general rules should be followed. The test head should be vented to the atmosphere unless there is a known pressure in the application. The lubricant level is normally midshaft

Table 13.6 Test Parameters--Automotive (Front Wheel Drive)

	Application						
	Axle		Engine		Transmission		
Test parameters	Driven axle	Dead axle	Front crank and Cam-shaft	Rear crank	Front pump	Output shaft	Manual input
Dynamic runout TIR, mm (in.)	0.25 (0.010)	0.25 (0.010)	0.25 (0.010)	0.25 (0.010)	0.25 (0.010)	0.41 (0.010)	0.25 (0.010)
Shaft-to- bore misalignment Offset, mm (in.)	0.13 (0.005)	0.25 (0.010)	0.25 (0.010)	0.13 (0.0050)	0.13 (0.005)	0.20 (0.008)	0.25
Direction of shaft rotation	Either	Either	Clockwise	Counter-clockwise	Clockwise	Either	Clock-wise
Hot test							
Reverse rotation	2 min at 200 rpm	2 min at 200 rpm	None	None	None 200 rpm	2 min at	None
Shaft speed, rpm	1200	1200	5000	5000	5000[b]	1200	5000
Temp., °C ± 3° (°F ± 5°)	104 (220)	82 (180)	141 (285)	141 (285)	149 (300)	Auto 121 (250) Manual 107 (225)	107 (225)
Duration, hr	350	350	350	350	400	350	350
Cold test (-40°C; °F) speed, rpm[c]	100	100	300	300	300	100	300

[a] TIR is double the value listed.

[b] Reciprocation at 10 strokes/min with an amplitude of 2.5 mm (0.10 in.) during shaft rotation.

[c] Rate of acceleration to be established by user and supplier.

Table 13.7 Test Parameters--Automotive (Rear Wheel Drive)

	Application						
	Axle		Engine		Transmission		
Test parameters	Pinion	Wheel	Front crank and Cam-shaft	Rear crank	Front pump	Rear	Manual input
Dynamic runout TIR,	0.25	0.25	0.25	0.25	0.25	0.25	0.25
mm (in.)	(0.010)	(0.010)	(0.010)	(0.010)	(0.010)	(0.010)	(0.010)
Shaft-to- bore	0.13	0.25	0.25	0.13	0.13	0.25	0.25
misalignment Offset[a], mm (in.)	(0.005)	(0.010)	(0.010)	(0.0050)	(0.005)	(0.010)	(0.010)
Direction of shaft rotation	Clockwise	Either	Clockwise	Counter-clockwise	Clockwise	Counter-clockwise	Clockwise
Hot test							
Reverse rotation	2 min at 600 rpm	2 min at 200 rpm	None	None	None	2 min at 600 rpm	None
Shaft speed, rpm	3600	1200	3600	3600	3600[b]	3600[c]	3600
Temp., °C ± 3°	100	66	121	121	149	Auto 121	(250)
107							
(°F ± 5°)	(212)	(150)	(250)	(250)	(300)	Manual 93 (200)	(225)
Duration, hr	350	350	350	350	400	350	350
Cold test (-40°C; -40°F) speed (rpm[d])	300	100	300	300	300	300	300

[a] TIR is double the value listed.

[b] Reciprocation at 10 strokes/mm with an amplitude of 2.5 mm (0.10 in.) during shaft rotation.

[c] Reciprocation at 15 strokes/mm with an amplitude of 12.7 mm (0.50 in.) during shaft rotation.

[d] Rate of acceleration to be established by and supplier.

Table 13.8 Test Parameters--Off-Highway Equipment and General

	Application						
		Engine		Axle			
Test parameter	Accessory drive	Front crank and cam-shaft	Rear crank	Pinion	Thru-shaft tandem pinion	Wheel	General
Dynamic runout TIR, mm (in.)	0.38 (0.015)	0.25 (0.010)	0.25 (0.010)	0.25 (0.010)	0.25 (0.010)	0.38 (0.015)	0.25 (0.010)
Shaft-to- bore misalignment Offset[a], mm (in.)	0.25 (0.010)	0.25 (0.010)	0.25 (0.010)	0.13 (0.005)	0.13 (0.005)	0.25 (0.010)	0.13 (0.005)
Direction of shaft rotation	Either	Clockwise	Counter-clockwise	Clockwise	Counter-clockwise	Either	Either
Hot test							
Reverse rotation	None	None	None	2 min at 300 rpm	2 min at 300 rpm	2 min at 60 rpm	None
Shaft speed, rpm	3000	3000	3000	3000	3000	1000	3000
Temp.°C± 3° (°F ± 5°)	121 (250)	135 (275)	135 (275)	93 (200)	93 (200)	82 (180)	107 (225)
Duration, hr	350	350	350	350	350	350	350
Cold test (-40°C;-40°F) speed, rpm[b]	300	200	200	150	150	60	200

[a] TIR is double the value listed.

[b] Rate of acceleration to be established by user and supplier.

except for engine seals. The sump level in the test machine is usually one-eighth of the shaft diameter to simulate the splash lubrication found in engine applications. Test shafts are hardened to 40RC and are plunge-ground to 0.25-0.50 μm (10-20 μin.) Ra with no machine lead. If wear sleeves are specified in the application, then they should be used in the functional tests. The seal lip and the test shafts should be prelubed lightly with the test lubricant unless the test parameters specify a completely dry condition. A minimum of six seals should be tested unless another number is specified by the test procedure. Leakage is collected in a vessel beneath the test head and is weighed daily to determine the leak rate. The failure criteria must be agreed upon by the seal user and supplier. If no agreement exists, then seal failure will occur when total leakage exceeds 5 g or if leakage exceeds 0.5 g in any 24 hr period.

Cold test parameters also appear on Tables 13.6-13.8 for various applications. The seal is installed in the test head, and the unit is cooled to the desired temperature. The test head, oil, and seal must remain at the temperature for a minimum of 4 hr before the spindle is started. The speed should build up to the maximum value at a uniform rate that is agreed upon by the seal user and supplier. The spindle is rotated for 10 min at the cold-temperature speed specification. The spindle speed is then increased to the value specified by the hot test and is run until the sump temperature

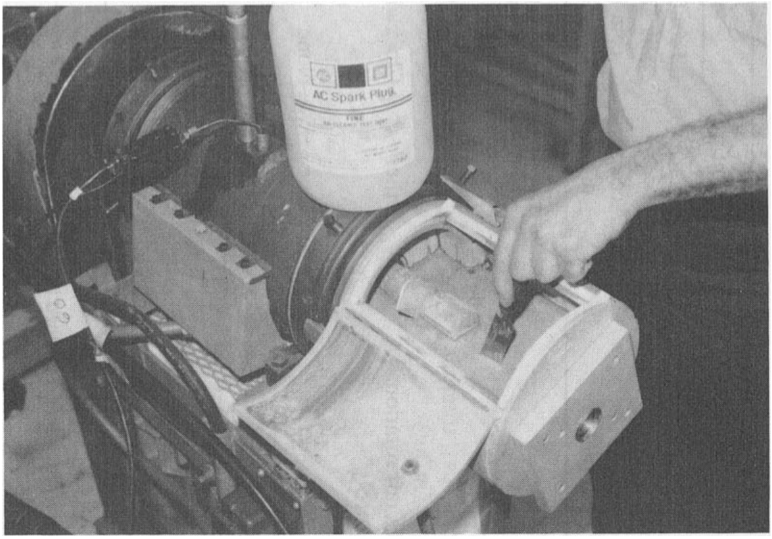


Fig. 13.23 Environmental test chamber. (Courtesy of Chicago Rawhide.)

reaches 20°C (68°F). The machine is stopped and cooled back to -40°C (-40°F). This cycle is carried out three times. Failure occurs if either seal fracture or leakage has occurred.

Environmental tests are conducted by attaching an environmental chamber (Fig. 13.23) to a standard test machine [167]. The chamber contains an impeller that is driven by an electric motor. The impeller is designed to lift dust, mud, or water, keep them in constant motion, and direct the contaminants toward the seal. The configuration will improve the consistency of the test results. The test media must also be specified and controlled to provide consistent test results (Table 13.9). A prescribed quantity of the specified contaminant is placed in the environmental chamber. The test oil or grease is placed in the test head and tested to the conditions recommended in Table 13.10. These conditions can be combined with any of the environmental test conditions outlined in Table 13.11. Failure occurs when oil leaks into the environmental chamber or when the contaminants leak into the oil.

13.19 SHAFT LEAD MEASUREMENT

Spiral grooves can be generated on a shaft surface by the relative axial movement of the finishing tool during the finishing operation. This condition, known as shaft lead, can have an effect on the sealing properties of a mating radial lip type of oil seal. Lead may be detected and quantified by partially wrapping a thread around a leveled shaft with a subtended relatively light weight and slowly rotating the shaft. If the thread traverses back and forth as the direction of rotation is reversed, shaft lead is present.

Table 13.9 Media for Environmental Test Chamber

Media	Composition
A	Distilled water
B	Salt water (1.03 specific gravity)
C	Equal parts of coarse, medium, and fine Arizona dust
D	Equal parts of aluminum oxide: 150, 240, and 400 grit
E[a]	Slurry: 1 part Wyoming bentonite 8 parts distilled water 1 part Arizona fine dust 1 part Arizona medium dust 1 part Arizona coarse dust

[a] Source of Arizona dust is AC Spark Plug Division of General Motors, Flint, Michigan.

Source: Ref. 167.

Table 13.10 Primary Lip Test Conditionsa

Parameter	Oil	Grease
Medium	SAE 10W30	Lithium
Temperature of medium	208°C (225°F)	70°C (175°F)
Level	Shaft centerline	Packed under seal lip
Shaft speed	8.6 m/sec (1700 ft/min)	2.0 m/sec (400 ft/min)
STBM	0.13 mm (0.005 in.)	0.25 mm (0.010 in.)
DRO	0.13 mm (0.005 in.) TIR	0.13 mm (0.005 in.) TIR
Test cycle	20 hr dynamic, 4 hr static	Continuous
Heat cycle	On only when running	Continuous
Duration	168 hr	168 hr
Failure criteria	Contaminant between or under lip(s); oil or grease leaking into contaminant	

[a] To be used with contaminant tests, Table 13.11.

STBM = shaft-to-bore misalignment; DRO = dynamic runout.

Source: Ref. 167.

The shaft can be checked for lead by chucking it in a lathe that can be reversed. Special test machines can be constructed with small reversible variable speed electric motors to drive the test piece (Fig. 13.24). In some cases, the unit is mounted on a comparator. The surface of the shaft and the thread appear greatly magnified on the screen of the comparator. This magnified view is helpful in determining the rate of thread advance when the shaft is rotated.

Several methods of preparing the shaft surface are recognized as acceptable for a seal running surface. The Rubber Manufacturers Association has described each method and rated the surfaces for use with shaft seals [150]. A rating of 5 is the best; 1 is the least desirable (Table 13.12). The RMA also describes a test to detect shaft lead.

The shaft or sleeve is mounted in the holding chuck and the system is checked to ensure that the shaft is level. The shaft or sleeve is lightly coated with a 5-10 centipoise viscosity silicone oil. A thread (Belding, Lily Article 3288, 100% extra strong quilting thread) of 0.23 mm (0.009 in.) diameter is draped over the surface of the shaft and a 28 g (1 oz) weight is attached to the loop at a distance below the shaft to create a string-to-shaft contact angle of 220-240°. Unwaxed dental floss or a 0.13 mm (0.005 in.) diameter monofilament line can be used in place of the quilting thread. The shaft is rotated at a speed of 60 ± 5 rpm, and the axial motion of the string is observed. The direction of shaft rotation is reversed, and the axial motion

Table 13.11 Contaminant Test Conditionsa

Environmental test conditions	Medium[b]	Medium level or application method	Chamber temp.	Impeller speed (rpm)	Test cycle
Dust test impeller	C,D	One-quarter of	Unheated	500	Impeller on during dynamic cycle
Water test impeller	A	One-quarter of	Unheated	500	Impeller on during dynamic cycle
Salt water test impeller	B	One quarter of	Unheated	500	Impeller on during dynamic cycle
Slurry test impeller	E	One quarter of	Unheated	500	Impeller on during dynamic cycle
Water drop test	A	5 cm^3/min on top of shaft	Unheated	500	Medium dropping during dynamic cycle
Salt water drop test	B	5 cm^3/min on top of shaft	Unheated	500	Medium dropping during dynamic cycle
Slurry drop test	E	5 cm^3/min on t top of shaft	Unheated	500	Medium dropping during dynamic cycle

[a] To be used with primary lip test conditions, Table 13.10.

[b] See Table 13.9.

Source: Ref. 167.

Table 13.12 Surface Preparation for Radial Lip Seals

Type of surface preparation	Description/comments	Rating[a]
Plunge grinding	The grinding wheel moves straight toward the surface (radially) rather than across the surface (axially), thus preventing spiral grooves that can promote leakage. For best results, use a clusterhead grinding tool. Work at a slow pace (3 in./min) and take shallow cuts (0.002 in. on the first pass and 0.001 in. on the second).	5--recommended procedure
Transverse grinding	The finish is produced on a centerless grinder, and the stock is pushed axially past the grinding wheel.	3--can produce lead if throughput is too high
Honing	Method produces a cross hatching pattern on the shaft surface. The finish is acceptable, but grooving acts as lead and allows lubricant to escape under the seal lip.	1
Paper polishing	Often done with hand-held emery cloth and will often produce lead. Method is effective if automatic equipment is used.	4

Glass bead/metal peening	Method produces a dimpled surface. Machine lead introduced by previous lathe work is almost invisible but still present. Metal shot can create kerfs at one side of the dimple that can act like machine lead.	4
Tumbling with aggregate	Method produces a uniform appearance, but it merely hides the machine lead tracks.	3
Roller burnishing	Method appears to flatten the surface peaks caused by lathe cutting. However, the original lead grooves are merely compressed, not removed.	3
Diamond burnishing	Method produces an accurate finish but does not remove machine lead. Instead, it creates even more lead because the burnishing medium moves axially.	1
Machine turning	Machine turning will almost always generate lead and should be followed with a subsequent finishing operation to remove lead.	1

[a] 5 is best; 1 is not preferred.

Source: Ref. 150.

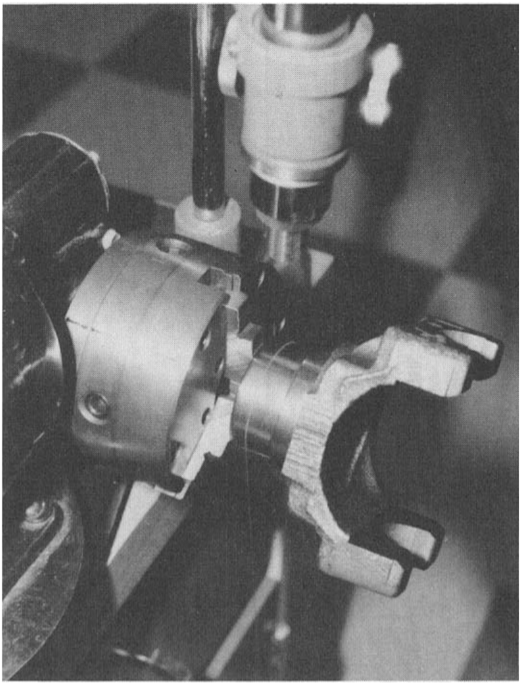


Fig. 13.24 Shaft lead detector. (Courtesy of Chicago Rawhide.)

of the string is observed once again. The interpretation of the observations appears in Table 13.13. The axial distance that the string advances during a specific time period can be used to compute the lead angle, which is the angle that the spiral grooves on the shaft make with the shaft centerline (Equation 13.14). The tolerance for lead angle is 0 ± 0.05°.

$$\text{Lead angle} = \arctan\left[\frac{\text{string advance}}{\pi D_s\ \text{(time of advance) (shaft rpm)}}\right] \tag{13.14}$$

13.20 MEASURING SHAFT SURFACE FINISH

The specification for the shaft surface finish is 0.2-0.50 µm (10-20 µin.) Ra. The finish is measured with a machine that has a very small stylus that can follow the shaft irregularities and convert the motion into electrical impulses that are recorded on a strip chart recorder. The stylus is placed on the surface of the shaft and is moved slowly across the surface (Fig. 13.25).

Table 13.13 Interpretation of Shaft Lead Test Results

Observation	Interpretation
Thread remains stationary for both directions of rotation at center and edges of the shaft.	No lead present.
Thread moves across the surface in the same direction for both directions of shaft rotation. Remounting the shaft end-for-end reverses the direction in which the thread moves across the shaft.	Tapered shaft--unable to determine if shaft lead is present.
Thread moves across the surface in the same direction for both directions of shaft rotation. Remounting the shaft end-for-end does not reverse the direction in which the thread moves across the shaft.	Shaft is not level--unable to determine if shaft lead is present.
Thread moves away from the center for both directions of shaft rotation.	Crowned shaft--unable to determine if shaft lead is present.
Thread moves toward the center for both directions of shaft rotation.	Cupped shaft--unable to determine if shaft lead is present.
The thread will advance from the fixed or arbor end of the shaft toward the free end when the shaft is rotating clockwise (CW). The thread will move from the free end of the shaft towards the fixed end for counter-clockwise (CCW) rotation.	Right-hand or CW lead.
The thread will advance from the fixed or arbor end of the shaft toward the free end when the shaft is rotating counterclockwise. The thread will move from the free end of the shaft toward the fixed end for clockwise rotation.	Left-hand or CCW lead.

Source: Ref. 150.

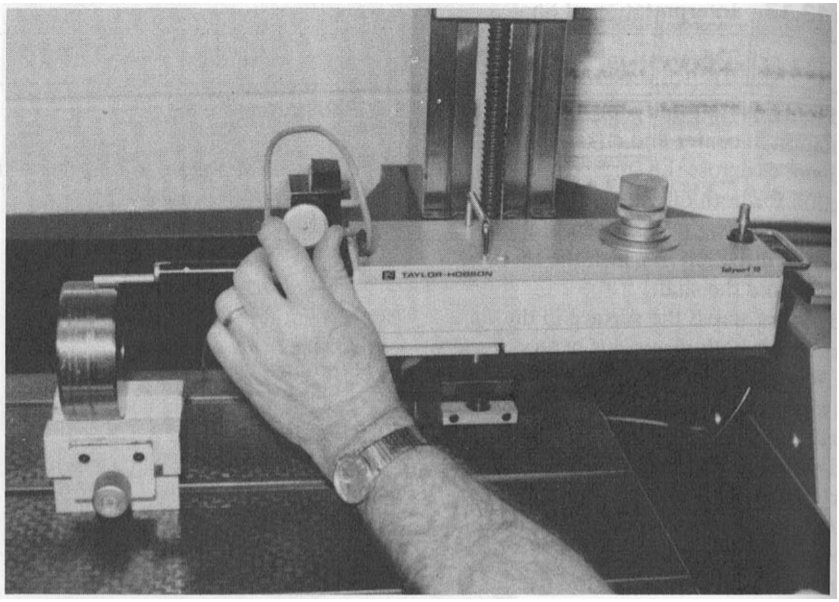

TAYLOR-HOBSON
Talysurf 10

Fig. 13.25 Measuring shaft surface finish. (Courtesy of Chicago Rawhide.)

APPENDIX A

Radial Lip Seal Terminology

Numbers in parentheses refer to item number in Fig. A.1

Air side The side of a seal that in normal use faces away from the fluid being sealed.

Air-side angle (4) The angle between the air-side surface and the shaft. Synonym: bar rel angle.

Air-side surface (5) The portion of the lip surface that faces the air side. Axial clearance *See*> Clearance, axial.

Axial dirt lip *See* Lip, axial dirt.

Barrel angle *See* Air-side angle.

Beam length *See* Lip length.

Bedding-in *See* Run-in.

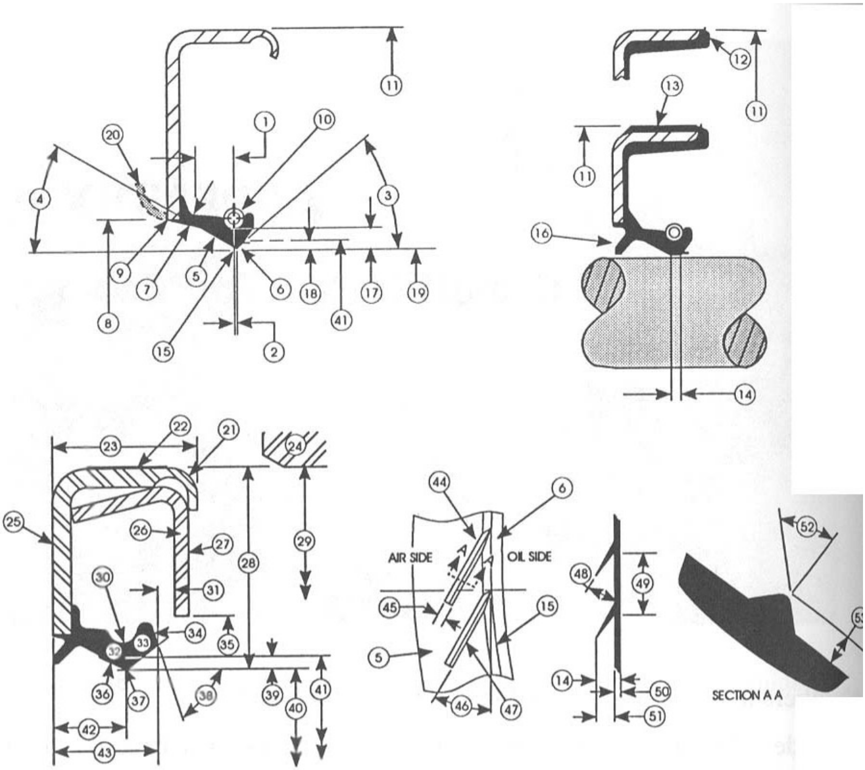
Bell mouth A condition where the contact between the sealing element and the shaft occurs on the air side of the seal and not on the seal tip.

Blister A raised cavity or sack that deforms a surface of the seal material.

Bond The adhesion established by vulcanization between two cured elastomer surfaces or between one cured elastomer surface and one nonelastomer surface.

Bore See Housing bore.

Cap The part of the seal head section that is removed during trimming or demolding.



1. Lip Length
2. "R" Value
3. Oil Side Angle
4. Air Side Angle
5. Air Side Surface
6. Oil Side Surface
7. Flex Thickness
8. Case I.D.
9. Heel Section
10. Garter Spring
11. Seal O.D.
12. Nose Gasket
13. Rubber O.D.
14. Contact Width
15. Contact Line
16. Radial Dirt Lip
17. Head Thickness
18. Sprung Interference
19. Lip I.D.
20. Axial Dirt Lip
21. Outer Case
22. O.D. Sealer
23. Case Width
24. Housing
25. Outside Face
26. Inner Case
27. Inside Face
28. Radial Wall Dimension
29. Housing Bore
30. Spring Groove
31. Axial Clearance
32. Head Section
33. Spring Retainer Lip
34. Toe Face
35. Inner Case I.D.
36. Rib (Helix Seal Only)
37. Static Lip
38. Molded Toe Angle
39. Unsprung Interference
40. Unsprung Lip Dia.
41. Shaft Diameter
42. Contact Line Height
43. Lip Height
44. Helix Rib
45. Rib Width
46. Helix Angle (Projected)
47. Rib Leading Face
48. Contact Angle
49. Rib Pitch
50. Static Lip Contact Width
51. Helix Contact Width
52. Rib Angle
53. Rib Height

Fig. A.1 Radial lip seal terminology. (Courtesy of Chicago Rawhide.)

Case, bonded A design feature of a type of radial lip seal wherein the heel of the sealing element is attached to the seal case by an adhesive during the molding process.

Case, clinched A design feature of a type of radial lip seal wherein the heel of the sealing element is attached to the case by clamping it between two convolutions (or folds) in the case.

Case, inner (26) A rigid cup-shaped component of a seal assembly, which is placed inside the outer seal case. It has one or more of the following functions: reinforcing member, shield, spring retainer, lip-clamping device.

Case, molded A design feature of a type of radial lip seal wherein the lip and the case are made integral in the molding process.

Case, outer (21) The outer thin-wall rigid structure of the lip seal assembly that contains the inner case, the primary seal ring, the spring parts, and the secondary seal.

Case, seal *See* Seal case.

Case I.D. (8) The smallest diameter of the seal case.

Cavity, mold A single unit or assembly of contoured parts in which a material such as an elastomer is shaped into a particular configuration.

Cavity, seal The annular area between a housing bore and a shaft, into which a seal is installed.

Checking Short axial cracks on the lip contact surface.

Clearance, axial (31) The gap between the toe face of the head section and the inside surface of the inner seal.

Cocked assembly An installation in which the plane of the outside seal face is not perpendicular to the shaft axis.

Coil One turn of the coiled wire garter spring.

Contact line (15) The circular line formed where the air-side and oil-side surfaces of the elastomeric lip element intersect. The contact line is a point when the seal element is viewed in cross section.

Contact line height (42) The axial distance from the outside seal face to the lip contact line.

Contact point The area where the seal lip contacts the shaft.

Contact width (14) The axial dimension of the contact area that results when the seal is installed on the shaft.

Contamination, surface Foreign matter on the seal surface.

Crack A sharp break or fissure in the sealing element.

Cut A deep discontinuity in the seal material whereby no material is removed.

Deformation A stress-induced change of form or shape.

Diameter, free-lip See Diameter, unsprung lip.

Diameter, functional lip The apparent inner diameter of the seal lip when the seal case is concentric with the outer diameter of the sizing mandrel in an air gauge, lightbox, or similar inspection equipment.

Diameter, molded lip See Diameter, unsprung lip.

Diameter, spring mean coil The spring coil diameter minus the spring wire diameter.

Diameter, spring outside coil The outer diameter of an individual helical coil of a garter spring.

Diameter, sprung lip See Lip I.D.

Diameter, trimmed lip The lip diameter in the free state (no spring) developed by knife trimming the molded portion of the sealing element to form the contact line.

Diameter, unsprung lip (40) The inner diameter of the seal lip, measured without the spring installed.

Dry running Operation of a seal without lubrication at the seal/shaft interface.

Eccentricity, lip I.D. to O.D. See Radial wall variation.

Eccentricity, shaft The radial distance by which the geometric center of a shaft is displaced from the axis of shaft rotation.

End play A measure of axial movement encountered or allowed, usually in reference to the shaft the seal lip contacts.

Extrusion Permanent displacement of part of a seal into a gap, under the action of fluid pressure.

Finish, shaft surface See Texture, shaft surface.

Flash Thin extensions of the elastomer formed by extrusion at the parting lines in the mold cavity or vent points.

Flashing A rapid change in fluid state, from liquid to gaseous. In a dynamic seal, this can occur when frictional energy is added to the fluid as the latter passes between the primary sealing faces or when fluid pressure is reduced below the vapor pressure of the fluid because of a pressure drop across the sealing faces.

Flex life The length of time to failure that indicates the relative ability of a material to withstand dynamic bending or flexing under specific test conditions.

Flex point Region where the seal lip will flex when the seal element is stretched over the shaft.

Flex thickness (7) The thickness of the region that flexes when the seal element is stretched over the shaft.

Flinger See Slinger.

Fluid side The side of the seal that in normal use faces the fluid being sealed.

Followability The ability of a seal lip to maintain a dam when the shaft has vibrations or dynamic runout.

Garter spring (10) A helically coiled spring with the ends connected to form a ring that is used to maintain the sealing force between the seal lip and the shaft.

Head section (32) The portion of a lip seal generally defined by the inside and outside lip surfaces and the spring groove.

Head thickness (17) The radial distance between the contact point and the bottom of the spring groove.

Heel section (9) The portion of the seal that is bonded to the case.

Helix angle (46) The angle between a helical rib and the lip line of contact.

Helix contact angle (48) The angle formed by the rib leading edge and the lip line of contact.

Helix rib See Rib.

Helix rib angle See Rib angle.

Helix rib height See Rib height.

Housing (24) A rigid structure that supports and locates the seal assembly with respect to the shaft.

Housing bore (29) A cylindrical surface that mates with the outside diameter of the outer seal case.

Hydroseal A sealing system with helically disposed elements formed on the shaft surface.

Inclusion Foreign matter included in the seal material.

Incomplete trim A trimmed surface that does not have all designated material removed.

Inner case See Case, inner.

Inner diameter, assembled spring The inner diameter of the garter spring, with the end securely joined.

Inner diameter, inner case (35) The inner diameter of the inner case of a radial lip seal.

Inside Face (27) The surface of the inner case that faces (and is usually in contact with) the fluid being sealed.

Interface The region between the static and dynamic sealing surfaces in which there is contact.

Interference (18) The difference between the shaft diameter and the sprung seal lip.

Knit line A blemish of the sealing element created by premature curing during molding operation.

Lead, shaft Spiral grooves on a shaft surface caused by relative axial movement of grinding wheel to shaft.

Leakage rate The quantity of fluid passing through a seal in a given length of time.

Length, deflected Refers to the working circumferential length measured on spring centerline of the garter spring with the seal lip assembled on a normal (designed) shaft diameter.

Length, spring free The total unconfined length of a spring. For a garter spring, it would not include the nib length.

Lip, axial dirt (20) A nonsprung axial lip at the heel of the elastomeric lip that impinges upon a radial flange and is used to exclude contaminants.

Lip force The radial force exerted by an extension spring and/or lip of a seal on the mating shaft. Lip force is expressed as force per unit of shaft circumference.

Lip height (43) The axial distance from the outside seal face to the toe face.

Lip I.D. (19) Primary sealing lip diameter that normally contains the spring.

Lip, molded A type of seal lip that requires no trimming to form the contact line.

Lip, primary The normally flexible elastomeric component of a lip seal assembly that rides against the rotating surface and effects the seal.

Lip, radial dirt (16) A nonsprung radial lip at the heel of the elastomeric lip that impinges upon the shaft and is used to exclude contaminants.

Lip, spring retainer (33) The portion of the primary lip that restricts the axial movement of the garter spring from a predetermined position.

Lip, static (37) That section of the helix seal lip incorporating the contact line.

Lip length (1) The axial distance from the flex point to the contact point. Also called beam length.

Load, radial The total force (load) acting on the seal lip that tends to maintain lip-shaft contact. It is the sum of the forces developed from seal interference and the garter spring.

LOP See Pressure, lip opening.

Lubricant, mold The substance used to coat the surfaces of a mold to prevent the elastomer from adhering to a mold cavity surface during vulcanization.

Lubricant starvation Lack of proper lubrication at the seal interface; may cause premature wear and early failure.

Mold cavity See Cavity, mold.

Mold impression A molded imperfection on the surface of the seal.

Molded toe angle (38) The angle between the toe face of a seal lip and the seal axis.

Nib See Spring nib.

Nick A void created in the seal material after molding or trimming.

Non-fill A void in the seal material.

Nose gasket (12) A portion of the elastomer that is attached to the face of the seal O.D. to form a gasket when the seal is installed and bottomed in the housing.

O.D. sealer See Sealer, case O.D.

Oil seal A seal designed primarily for the retention of oil.

Offset The radial distance between the axis of the seal bore and axis of shaft rotation. Synonym: shaft-to-bore misalignment (STBM).

Oil side The side of a seal that in normal use faces the fluid being sealed.

Oil-side angle (3) The angle between the oil side surface and the shaft.

Oil-side surface (6) The portion of the lip surface that faces the oil side of the application.

Outer case See Case, outer.

Out-of-round, shaft The deviation of the shaft cross section from a true circle. Out-of-round is measured as the radial distance, on a polar chart recording, between concentric circumscribed and inscribed circles that just contain the trace and are so centered that the radial distance is minimized.

Outside face (25) The seal case surface (perpendicular to the shaft axis) that is not in contact with the fluid being sealed.

Pitch, helix seal rib (49) The circumferential displacement between adjacent helical ribs of a lip seal. Synonym: Rib pitch.

Plunge ground The surface texture of a shaft or wear sleeve produced by presenting the grinding wheel perpendicular to the rotating shaft with out axial motion.

Porosity A multitude of minute cavities in the seal material.

Pressure, contact The average pressure exerted by a seal on a shaft. This pressure is computed by dividing the total lip force by the total lip contact area. Sometimes referred to as radial pressure.

Pressure, lip opening (LOP) The air-side pressure necessary to maintain flowing air at 10,000 cm^3/m between the contact surface of a radial lip seal and a shaft-size mandrel.

Pressure, seal cavity The pressure of a fluid being sealed.

Pressure, spring The contact pressure that results from the spring load.

PV factor An arbitrary term that is the product of face pressure and relative sliding velocity. The term is normally considered to provide some measure of severity of service or seal life.

Radial dirt lip See Lip, radial dirt.

Radial wall dimension (28) The distance between the seal lip contact line and the seal outside diameter measured in a radial direction on a finished seal in the free state.

Rib (36, 44) A long, narrow projection that is normally triangular in cross section and is molded into the outside lip surface of a helix seal. It is oriented at an angle to the shaft axis. One end of the rib forms part of the seal-lip contact surface.

Rib angle (52) The angle formed by the leading edge of the rib and a line perpendicular to a plane tangent to the outside lip surface at the centerline of the rib base. Synonym: helix rib angle.

Rib height (53) The height of the helical ribs, measured perpendicular to the outside lip surface. Synonym: helix rib height.

Rib leading face (47) The face of the helix seal rib that is closest to the fluid side of the seal.

Rib pitch See Pitch, helix seal rib.

Rib width See Width, helix rib.

Roughness, axial surface Surface roughness of a shaft measured in a direction parallel with the centerline axis.

Roughness, circumferential surface Surface roughness of a shaft measured in a direction (plane) normal to the centerline axis.

Rough trim A trimmed surface with irregularities on the outside and inside lip surfaces in the immediate vicinity of the contact line.

Rubber O.D. (13) A portion of the elastomer that is attached to the outside surface of the case.

Run-in The period of initial operation during which the seal lip wear rate is greatest and the contact surface is developed. Synonym: bedding-in.

Runout, dynamic Twice the distance the center of the shaft is displaced from the center of the rotation and expressed in TIR. The runout to which the seal lip is subjected due to the outside diameter of the shaft not rotating in a true circle. Synonym: shaft runout.

R value (2) The axial distance between the spring centerline and the contact point; can be measured in the free state or on shaft.

RWV See Variation, radial wall.

Scoop trim A trimmed surface that is concave.

Scratch A shallow discontinuity in the seal material from which no material has been removed.

Scuffing Metal surface degradation resulting from adhesive wear.

Seal, birotational A rotary shaft seal that will seal fluid regardless of the direction of shaft rotation.

Sealing capacity The difference in leakage rates of a hydrodynamic seal and a nonhydrodynamic seal when tested on a shaft with a spiral groove that tends to pump oil out of the sump when the shaft is rotating.

Seal case A rigid member to which the seal lip is attached.

Seal cavity See Cavity, seal.

Seal, dynamic A seal that has rotating, oscillating, or reciprocating motion between it and its mating surface, in contrast to stationary seals such as gaskets.

Sealer, case O.D. (22) A coating applied to the case O.D. to prevent leakage between the seal case and the housing bore.

Seal, helix An elastomeric hydrodynamic lip seal having helical ribs on the outside lip surface.

Seal, hydrodynamic A dynamic sealing device with ribs, triangles, wavy lips, etc. that utilizes the viscous shear and inertial forces of the fluid to generate a pressure differential that opposes fluid flow.

Seal, lip An elastomeric seal that prevents leakage in dynamic and static applications by reason of controlled interference between the seal lip and the mating surface.

Seal, mechanical Any material or device that prevents or controls the passage of matter across the separable members of a mechanical assembly.

Seal, radial A seal that exerts radial sealing pressure in order to retain fluids and/or exclude foreign matter.

Seal, radial lip A type of seal that features a flexible sealing member referred to as a lip. The lip is usually of an elastomeric material. It exerts radial sealing pressure on a mating shaft in order to retain fluids and/or exclude foreign matter.

Seal slant The difference between the maximum and minimum axial dimensions from the seal lip contact line to the outside face of the case.

Seal, split A seal whose primary sealing element is split approximately parallel with the shaft axis centerline. Typically used where conventional installation methods are impractical or impossible.

Seal, unirotational A seal designed for applications requiring a single direction of shaft rotation.

Seal, unitized A seal assembly in which all components necessary for accomplishing the complete sealing function are retained in a single package.

Seal tip The point where the air and oil side surfaces intersect.

Shaft-to-bore misalignment (STBM) See Offset.

Shaft runout See Runout, dynamic.

Sleeve See Wear sleeve.

Slinger A washer-like device used for imparting radial momentum to a liquid in order to keep the latter away from the sealing interface. Often incorporated into a wear sleeve. Also called flinger.

Slip-stick A friction-related phenomenon in which the sealing element tends to momentarily adhere and rotate with the shaft surface until the elastic characteristics of the sealing element overcome the adhesive force, causing the seal lip to lose contact with the rotating shaft long enough to allow leakage. This cycle repeats itself continuously and is normally associated with nonlubricated and boundary-lubricated conditions.

Speed, surface The linear velocity calculated from the shaft rotational speed using the nominal shaft diameter.

Spiral trim A trimmed surface that has a spiral pattern.

Spring, finger A spring consisting of a multiple number of cantilevered elements located circumferentially on a ring. It can be designed to produce either a radial or an axial force.

Spring groove (30) A depression formed in the head section of the seal. It is generally semicircular in form and serves to accommodate and locate the garter spring.

Spring index The radio of the mean coil diameter to the wire diameter of a garter spring.

Spring nib A short end section of an extension spring formed by a reduction in the coil diameter used to join the two ends in forming a garter spring.

Spring rate The force, independent of initial tension, that is required to extend the working length of a spring a unit distance.

Spring retainer lip See Lip, spring retainer.

Spring wind-up The tendency of a garter spring with its ends assembled together to deform from a flat surface. Excessive spring wind-up results in the spring forming a figure 8 configuration.

Stability, dimensional The ability to retain manufactured shape and size after having experienced the combination of operating stresses and temperatures.

Static lip See Lip, static.

STBM See Offset.

Step trim A trimmed surface having a discontinuity perpendicular to the contact line.

Static lip contact width See Width, static lip contact.

Sump temperature The temperature of the fluid contained within the machinery sump.

Surface, air-side See Air-side surface.

Surface, contact The portion of the seal lip that circumferentially contacts the shaft to form the seal/shaft interface.

Surface, oil-side See Oil-side surface.

Surface, trimmed seal The lip surface formed by a knife cutting operation to develop the contact line.

Tear The removal or separation of a portion of the sealing element.

Tension, initial spring The”preload” that has been wound into the coils of a spring during the coiling operation.

Test, accelerated life Any set of test conditions designed to reproduce, in a short time, the effects obtained under service conditions.

Test, bench A laboratory test in which the functional operating conditions are approximated, but the equipment is conventional laboratory equipment and not necessarily identical with that in which the product will be used.

Test, field A test performed in the actual environment in which the product will be used.

Test, flex A laboratory method used to evaluate the resistance of a material to repeated bending.

Test, life A laboratory procedure used to determine the period of operation that a component or assembly will operate until it no longer performs its intended function.

Texture, shaft surface A term used to describe the quality, appearance, or characteristic of the shaft surface resulting from operations, such as grinding, polishing, burnishing, and so on.

Thickness, film In a dynamic seal, the distance separating the two surfaces that form the primary seal.

Toe face (34) The annular surface of the spring retaining lip. Also called molded toe face.

Trim The removal of superfluous parts from a molded product, usually removal of parting line flash or feed sprues.

Trim, crooked Improper trimming during the manufacturing process. The plane that contains the contact surface is not perpendicular to the shaft axis. This condition is sometimes referred to as seal slant.

Trim cut Damage to the elastomeric portion of the seal during trimming.

Trim, rough Irregularities on the outside and inside lip surfaces in the immediate vicinity of the contact line.

Unbonded flash Flash that does not properly adhere to the mating material to which it is intended to be bonded.

Underlip temperature The temperature of the oil between the rotating shaft and the seal lip in the contact region.

Underlip temperature rise The difference between the underlip and sump temperature.

Unsprung interference (39) The difference between the shaft diameter and the unsprung lip diameter.

Variation, contact line height The difference in the contact line height as measured at any two points on the contact line. Maximum contact line height variation is defined as seal slant.

Variation, radial wall (RWV) The difference between the minimum and maximum radial wall dimensions when measured around 360° of the seal lip. Synonym: Lip I.D. to O.D. Eccentricity.

Vibration, torsional A vibration that has a circumferential or angular direction. It is often generated by a stick-slip action between mating seal faces.

Wear sleeve A replaceable metal ring generally used in assemblies to avoid expensive shaft replacement caused by grooving that may occur at the seal/shaft interface.

Weepage A minute amount of liquid leakage by a seal.

Wetting A formation of a continuous film of a liquid on a surface.

Width, case (23) The total axial width of the seal case.

Width, helix contact (50) The axial width of the portion of the contact surface of a helix seal that is formed by the helical ribs. It is equal to the total axial width of the contact surface minus the width of the static lip.

Width, helix rib (45) The maximum width of a helical rib measured perpendicular to the longitudinal axis of the rib.

Width, static lip contact (51) The axial width of the contact surface developed by a static lip.

Young's modulus The ratio of the stress to the resulting strain (the latter expressed as a fraction of the original height or thickness in the direction of the force).

APPENDIX B
Elastomer Terminology

Accelerator A substance that hastens the vulcanization of an elastomer, allowing it to cure in a shorter period of time or at a lower temperature.

Accelerator activator A substance that hastens the rate of vulcanization by activating the accelerator.

Aftercure Continuation of vulcanization after the desired cure is effected and the heat source is removed.

Antioxidant An organic substance that inhibits or retards oxidation.

Antiozonant A substance that retards or prevents the formation of cracks due to the deleterious affect of ozone when an elastomer is exposed to it while in tension, either statically or dynamically.

Bleeding Migration of plasticizers, waxes, or similar materials to the rubber surface, forming a film or beading.

Bloom A dusty- or milky-looking deposit that appears on the surface of a part after molding and storage, caused by the migration of an organic liquid or solid to the surface.

Buna N See> Nitrile.

Compound A term applied to a mixture of polymers and other ingredients to produce a usable rubberlike material.

Compression set Typically measured as the percent not recovered of a test specimen that has been compressed to 75% (25% compression) of its original height and held in a fixture in this condition for a specified time and temperature. A measure of the permanent deformation of an elastomer under load for time at a specified temperature.

Copolymer A polymer consisting of two different monomers chemically combined.

Creep The time-dependent part of a strain resulting from stress.

Cure time The time required to produce vulcanization at a given temperature.

Curing temperature The temperature at which an elastomeric product is vulcanized.

Durometer An instrument that measures the hardness of rubber by the penetration (without puncturing) of an indenter point into the surface of the rubber.

Durometer hardness An arbitrary numerical value that indicates the resistance to penetration of the indenter point into the rubber surface. Value may be taken immediately or after a very short specified time.

Elasticity The property of a material that causes it to return to its original shape after deformation.

Elastomer An elastic rubberlike substance, such as natural or synthetic rubber.

Elongation The increase in length of a specimen due to a tensile force; expressed as a percentage of the original specimen length.

Filler A solid compounding ingredient that may be added, to a polymer, usually in finely divided form and in relatively large proportions.

Flexibility, cold Flexibility of a material during exposure to a predetermined low temperature for a specific length of time.

Fluoroelastomer A saturated polymer in which hydrogen atoms have been replaced with fluorine. Characterized by excellent chemical and heat resistance.

Hardness The resistance to indentation. Measured by the relative resistance of the material to an indenter point of any one of a number of standard hardness testing instruments. See also Durometer hardness; Shore hardness.

International rubber hardness degree (IRHD) A standard unit used to indicate the relative hardness of elastomeric materials, where zero represents a material having a Young’s modulus of zero, and 100 represents a material of infinite Young’s modulus.

Modulus, rubber The tensile stress of a rubber at a specified elongation. A measure of resistance to deformation.

Modulus, Young’s See Young’s modulus in Appendix A.

Monomer A chemical compound composed of a single type of organic molecule and capable of linking with other types of monomers to form a polymer.

Mooney scorch The rate at which a rubber compound will cure or vulcanize measured by means of a Mooney viscometer test instrument.

Mooney viscosity The plasticity or viscosity of an uncompounded or compounded unvulcanized elastomeric material measured by means of a Mooney shearing disk viscometer.

Nitrile A general term for a copolymer of butadiene and acrylonitrile.

Oil resistance The measure of the ability of an elastomer to withstand the deteriorating effect of oil on mechanical properties.

Oil swell The change in volume of a rubber material due to absorption of oil.

Optimum cure State of vulcanization at which the most desirable combination of properties is attained.

Outgassing A vacuum phenomenon wherein a substance spontaneously releases volatile constituents in the form of vapors or gases. In rubber compounds, these constituents may include water vapor, plasticizers, air inhibitors, etc.

Overcure A degree of cure greater than the optimum, causing some desirable properties to be degraded.

Oxidation The reaction of a compound with oxygen, usually detected by a change in feel or appearance and/or a change in physical properties.

Permanent set The residual unrecoverable deformation in an elastomeric part after the load causing the deformation has been removed.

PHR Parts per hundred rubber. All ingredients of a compound are given in amounts based on the total amount of the base polymer or master batch.

Plasticity The degree or rate at which unvulcanized elastomer and elastomeric compounds will flow when subjected to forces of compression, shear, or extrusion.

Plasticizer A material that when incorporated in an elastomer or a polymer, will change its hardness, flexibility, processability, and/or plasticity.

Polyacrylate A type of elastomer that is characterized by an unsaturated chain and is a copolymer of an alkyl acrylate and some other monomer such as chloroethyl vinyl ether or chloroacetate.

Polymer Generic term for an organic compound of high molecular weight that consists of recurrent structural groups (see Monomer).

Polymerization The process undergone by certain organic compounds when they react together to form a single molecule of higher atomic weight.

Postcure A secondary curing cycle after vulcanization to provide for stabilization of the part and drive off decomposition products formed in the vulcanizing process.

Precure The first cure of a material that is given more than one cure in its manufacture. Synonym: partial cure.

Reinforcing agent A material, commonly clay, carbon black, or fine silica, that is dispersed in an elastomer to enhance its physical properties.

Relaxation, stress See Stress relaxation.

Resilience In elastomeric or rubberlike materials subjected to and relieved of stress, the ratio of energy given up on recovery from the deformation to the energy required to produce the deformation. For an elastomer, usually expressed in percent.

Resistance, cold The ability of a seal or sealing material to withstand the effects of a low-temperature environment without loss of serviceability.

Resistance, heat The ability of a seal or sealing material to resist the deteriorating effects of elevated temperatures.

Resistance, ozone The ability of a material to withstand the deteriorating effects of ozone (surface cracking).

Resistance, tear The property of an elastomeric material to resist tearing forces.

Rheometer An instrument designed to measure the curing characteristics of an elastomer specimen heated and maintained under continuous pressure during vulcanization via the sinusoidal oscillation of a conical disk. The force (torque) required to oscillate the disk is directly proportional to the stiffness (shear modulus) of the specimen. As the modulus increases with vulcanization, the torque is plotted against a time scale to give a cure curve of stiffness vs. time.

Scorching Premature curing of an elastomeric compound prior to obtaining the desired properties during normal vulcanization.

Set See Compression set; Permanent set.

Shore hardness The relative hardness of an elastomer obtained by use of a Shore durometer instrument.

Silicone rubber A type of elastomer having a basic polymer of dimethyl polysiloxene with various attached vinyl or phenyl groups.

Stress relaxation A characteristic of an elastomer wherein a gradual increase in deformation is experienced under constant load after the initial deformation.

Synthetic rubber A synthetic elastomer made by polymerization of one or more monomers.

Tensile strength, ultimate The force per unit or original cross-sectional area at the moment of specimen rupture.

Terpolymer A polymer consisting of three different monomers chemically combined.

TR Temperature retraction test. Measures the low-temperature properties of an elastomer by stretching a specimen 50%, freezing it at a temperature below its crystallization temperature, then allowing the stretched specimen to warm gradually, measuring the temperature at various percent recovery values. For example, TR-10 is the temperature in degrees Celsius or Fahrenheit at which the specimen has retracted 10% of its stretched length (90% of original stretch). This temperature is the lowest at which an elastomer still exhibits rubberlike properties and is a measure of maximum low-temperature sealability. TR-70, the temperature at 70% retraction (30% original stretch), is a good measure of low-temperature compression set.

Ultimate elongation Percent increase over the original dimension at break or rupture of the test specimen.

Undercure Degree of cure less than the desired optimum; may be evidence by tackiness, loginess, or inferior physical properties.

Volume swell A measure of the percent increase or decrease in volume of a test specimen subjected to immersion in a test fluid.

Vulcanization An irreversible process during which a rubber compound, through a change in its chemical structure (e.g., cross-linking) becomes less plastic and more resistant to swelling by organic liquids and the elastic properties are confined, improved, or extended over a greater range of temperature.

Vulcanizing agent A material, commonly a sulfur compound or peroxide, that produces vulcanization of an elastomeric compound by means of cross-linking the polymer chains.

APPENDIX C

Lubricant Terminology

Abrasive wear Removal of materials from surfaces in relative motion by the cutting or abrasive action of a hard particle (usually a contaminant).

Absolute viscosity Fundamental measure of viscosity, measured in a capillary tube viscometer under constant temperature and pressure. The unit of measure is the poise (P), where 1 P = 1 (dyn-sec)/cm^2, but viscosity is usually reported in centipoise (cP) (1 P = 100 cP. See Fig. C.1.

Absorption The process by which one substance draws another substance into itself, e.g., a sponge absorbing water.

Acid A compound containing hydrogen that can be replaced by a metal, forming a salt. In grease manufacture, fatty acids are used, which form a specific type of salt known as a soap (see Soap).

Activated The condition of a substance that has been treated in a special manner to increase its activity, which may be physical, such as decolorizing or absorption, or chemical, such as the behavior of catalysts. Bentonite and silica are materials that receive special treatments of this type when they are used as inorganic grease thickeners.

Additive Any material added to a lubricating grease or lubricating oil to improve its suitability for service. It may improve a property already possessed by the lubricant or give it properties it does not naturally possess. (Typical examples are antioxidants and “E.P.” or anti-weld additives.)

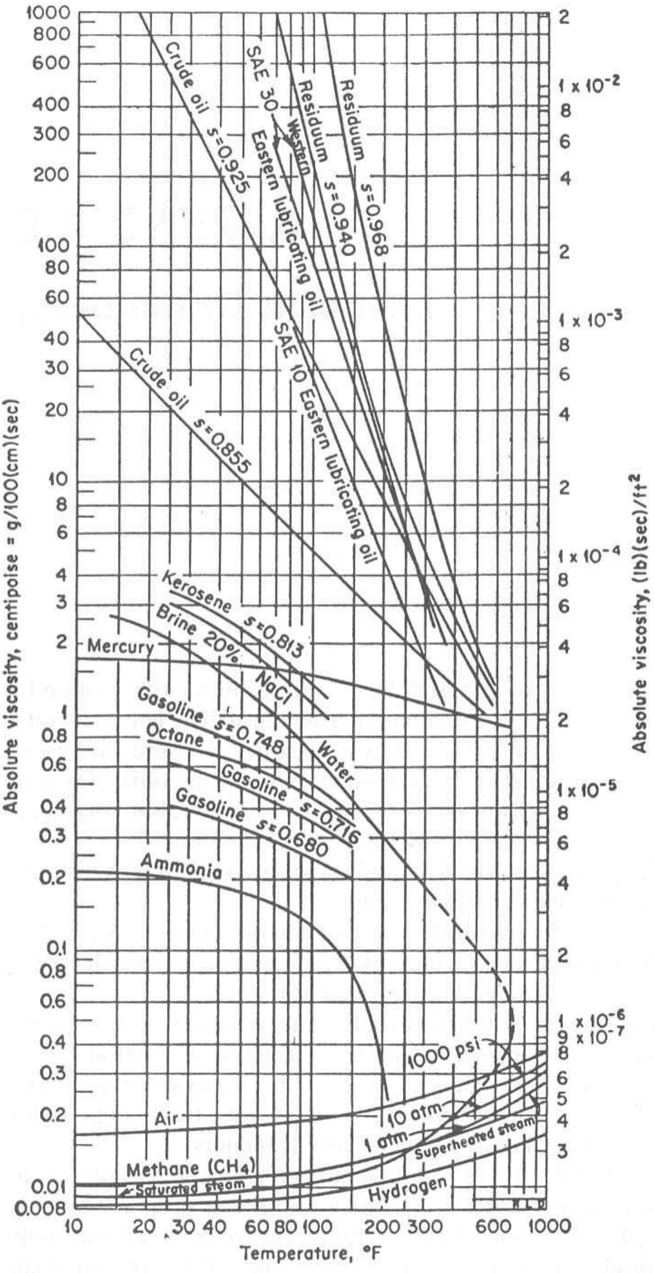


Fig. C.1 Absolute viscosity versus temperature. [Courtesy of McGraw-Hill *Fluid Mechanics Handbook* (1968).]

Adhesion The force or forces causing two materials, such as a lubricating grease and a metal, to stick together.

Adhesive wear Removal of materials from surfaces in relative motion as a result of surface contact. Galling and scuffing are extreme forces.

Age hardening Increase in consistency (hardening) of a lubricating grease with storage time.

AGMA American Gear Manufacturers Association.

AGMA lubricant numbers Arbitrary numbers assigned by AGMA to designate various viscosity grades of gear lubricants.

Alkali Any substance having basic properties (as contrasted with acid properties). In a restricted sense in grease manufacture, the hydroxide of an alkali metal, such as lithium, sodium, calcium, or potassium.

Aluminum base grease A grease prepared from a lubricating fluid thickened with aluminum soap.

Amorphous Without form. A term applied to such materials as carbon black, as contrasted with such crystalline materials as graphite or molybdenum disulfide.

Anhydrous Devoid of water.

Aniline point The lowest temperature at which a fluid is completely miscible with an equal volume of freshly distilled aniline. A measure of the solvent quality of a petroleum product that provides a clue as to its type--paraffinic, naphthenic, etc. It also indicates the material’s tendency to cause rubber seals to swell.

Antifoam additive An additive that causes air bubbles to break on the surface and prevents buildup of foam. Also acts as an antioxidant (cannot prevent air entrainment, but functions as an aid to quicker release of entrained air).

Antioxidant Oxidation inhibitor. An additive, usually incorporated in a relatively small proportion, that retards oxidation of lubricants, including greases and gear lubricants.

Antirust or anticorrosion additive An additive that helps prevent oxidation of metal by displacing water from metal surfaces. It plates to metal with a polarized effect to give an internal “umbrella,” helping to resist the formation of iron oxide (rust).

API American Petroleum Institute.

API engine service classification A system of letter designations defining broad classes of engine services.

Apparent viscosity The ratio of shear stress to rate of shear of a non-Newtonian fluid, calculated from Poiseuille’s equation and measured in poises. The apparent viscosity of most greases varies with changing rates of shear and temperature and must therefore be reported as the value at a given shear rate and temperature.

Appearance Those characteristics of a lubricating grease that are observable by visual inspection only. This general term includes various characteristics described under Bloom, Bulk appearance, Color, Luster, and Texture.

Ash A measure of the amount of inorganic material in a lubricating oil. Determined by burning the oil and weighing the residue. Results expressed as percent by weight.

ASTL American Society of Tribology and Lubrication (formerly called ASLE).

ASTM American Society for Testing and Materials.

Barium-base grease A grease prepared from a lubricating fluid thickened with barium soap.

Base A refined mineral oil, free of additives, used as a component in a lubricant blend.

Basic barium compounds Compounds of barium in which the barium is not completely reacted. These compounds show a basic reaction and assist in balancing certain grease formulations.

Bentonite A naturally occurring earth or clay. In a treated form it is sometimes used as a grease thickener and produces nonmelting grease.

BIA Boating Industry Association

Bleeding The separation of oil from a grease structure. A certain amount of bleeding is considered desirable in greases, because it tends to provide continuous oil lubrication to bearings.

Blending The process of mixing lubricant components for the purpose of obtaining desired physical and/or chemical properties. See also Compounding.

Bloom The surface color (usually blue or green) of a lubricating oil or grease when viewed by reflected daylight at an angle of about 45° from the surface.

Bloom is associated with the absorption of ultraviolet light in the oil and may not be visible if the sample is viewed by artificial light.

Blow-by Gases and unburned fuel that bypass the compression rings and go to the crankcase each time the engine fires.

Boundary lubrication A state of lubrication that occurs when, due to speed, load, or lubricant combination, the “thick film” or hydrodynamic conditions do not completely separate the rubbing surfaces. Special additives are sometimes used for bearing protection under these conditions.

Bright stock High-viscosity refined lubricating oils used as a component in a lubricant blend.

Bulk appearance Visual appearance of grease when the undisturbed surface is viewed in an opaque container. Bulk appearance should be described in the following terms:

Smooth A surface relatively free of irregularities.

Rough A surface composed of many small irregularities.

Grainy A surface composed of small granules or lumps of constituent soap particles.

Cracked Showing surface cracks of appreciable magnitude. In describing such a lubricating grease, the number and size of the cracks should be included in the description.

Bleeding Showing free oil on the surface of the lubricating grease or in the cracks of a cracked grease.

Calcium-base grease A grease made from a lubricating fluid thickener with calcium soap.

Carbon dioxide (CO_2) A colorless, odorless gas produced by complete combustion of a hydrocarbon fuel-air mixture. A component of the exhaust from an internal com bustion engine.

Carbon monoxide (CO) A colorless, highly toxic gas produced by the combustion of organic compounds with a limited amount of oxygen. A component of automotive engine exhaust now controlled and limited in amount by federal and state standards for clean air.

Catalyst A material that promotes some chemical action without itself being consumed in the reaction.

Cavitation In a lubricating grease dispensing system, failure of the material to flow to the suction of the system pump for any reason. See also Feedability.

Centipoise and centistoke A centipoise (cP) is 1/100 of the unit of absolute viscosity (the poise); e.g. the viscosity of water at 20°C is approximately 1 cP. The centipoise is derived from one kinematic unit of viscosity, the centistoke (cSt), by multiplying the latter by the density of the liquid, i.e. 1 cP = 1 cSx density of the liquid. These units are part of the metric system, commonly used in Europe and becoming adopted in the United States and Canada. See Appendix D.

Centralized lubrication A system of lubrication in which the lubricant for var ious parts of a machine or a group of machines is supplied from a central location.

Channeling (1) A term used in connection with lubricating greases to describe the (usually desirable) tendency to form a channel by the working down of lubricating grease in a bearing, leaving shoulders of unworked grease that serve as seal and reservoir. (2) A term used in connection with liquid lubri cants and flow-type lubricating greases to describe the tendency at low tem peratures for these materials to form a plastic structure sufficiently strong to resist flow under gravitational forces only. (Similar but not identical to the pour point of liquid lubricants, it is measured by empirical tests such as Federal Method 3456-T.)

Coefficient of friction The ratio of the friction between two surfaces to the pressure between them. A low coefficient of friction means low friction losses, influenced by the viscosity and character of the lubricant and by materials, surface conditions, and other factors.

Cohesion Molecular attraction causing particles of a substance to attract each other and to stick together. Cohesion contributes to the resistance of a lubricating grease to flow.

Cold sett See Sett grease.

Colloid A substance composed of particles of a size larger than molecules but small enough to possess reasonable dispersion stability in a two-phase system. A lubricating grease is a colloidal system. See also Thickening agent.

Color (of lubricating grease) The shade and intensity that show when lubricating grease is viewed under conditions that eliminate bloom. These conditions may be obtained by viewing the lubricating grease in an opaque container such as a metal package, by reflected light, from a position approximately perpendicular to the surface or by viewing it with transmitted light after placing the sample on a transparent plate. When colors of lubricating grease are referred to, the method by which these colors are determined should be clearly indicated. Colors of lubricating greases are best described in terms of the predominant hue such as amber or brown (or perhaps green, red, or blue for dyed greases) with a qualifying adjective describing intensity in terms of light, medium, or dark.

Color (of lubricating oil) The shade that shows when oil is viewed under transmitted light. Usually lubricating oil colors are obtained by viewing the oil under specified conditions in test equipment. Several such methods are available, the most widely used being ASTM D-1500-64, which describes the color in terms of numbers 1--8, with higher numbers being darkest.

Complex soap A soap in which the soap crystal or fiber is formed by cocrys talization of two compounds: (1) the normal soap (such as metallic stearate or oleate) and (2) the complexing agent. Examples of complexing agents are the metallic salts of short-chain organic acids such as acetic or lactic or the inorganic salts such as carbonates or chlorides. Another complexing agent is water, in which case the soap may also be referred to as a hydrated soap.

Compounding The mixing or combining of lubricant components with other components for the purpose of securing chemical and/or physical properties not usually obtainable by the blending of fluid lubricant components alone. (See Blending).

Compounded oil Mineral oil to which has been added vegetable or ani mal oil or chemical lubricating oil additives to enhance certain physical or chemical properties of the finished blend.

Conradson carbon The percentage by weight of material remaining after the oil has been burned according to standard conditions.

Consistency (hardness) The degree to which a plastic material such as lubricating grease resists deformation under the application of force. It is therefore a characteristic of plasticity, as viscosity is a characteristic of fluidity. Consistency is usually indicated by either apparent viscosity or ASTM penetration.

Corrosion Chemical attack or action on a metal by an acid, alkali, oxygen, chlorine, sulfur, or other chemical. Distinct from metal destruction by wear and may be evident as either discoloration or pitting.

Corrosion inhibitor An additive that protects alloy bearings, such as copper-lead, cadmium-nickel, and ferrous metals (cylinder walls and valves), against attack by acids formed in an engine.

Corrosive wear Removal of materials by chemical action.

Cup grease Usually a calcium-base grease having a yellowish color and a buttery structure. For use in grease cups.

Deposits Oil-insoluble materials that result from oxidation of the oil and contamination from external sources (products of combustion in the internal combustion engine) and settle out on machine or engine parts as sludge and varnish.

Detergent Lubricating oil additive, usually an organometallic chemical, that controls deposit formation in machines and internal combustion engines by keeping oil-insoluble materials suspended in the oil, thus permitting removal by draining.

Dispensability The property of a grease that governs the ease with which it can be transferred from its container to its point of application. The term is used mostly in discussion of dispensing in grease systems, where it includes the properties of both pumpability and feedability.

Dispersant A lubricating oil additive, usually an organic compound containing nitrogen groups, that controls deposit formation by keeping deposit precursors finely dispersed in the oil so they cannot form oil-insoluble materials that can settle out of the oil.

Dropping point The temperature at which a drop of material falls from the orifice of the test apparatus under the conditions of ASTM test D566-42. (This test should be considered as having very limited bearing upon service performance.) It is not the melting point of the grease, which is a term of little or no significance when applied to plastic materials such as lubricating grease. These materials are characterized by a melting range during which the material becomes steadily softer. Only through the establishment of an arbitrary and fully controlled test procedure such as ASTM Dropping Point, can any single temperature be established as a characteristic of the material. A similar test (differing only in details), the IP “drop point” usually gives a lower value.

Drum A container with a capacity of 55 U.S. gallons.

Dry-film lubricant Any of a class of lubricants wherein the reduction of friction and wear during sliding is caused by making the shearing take place within the crystal structure of a material with low shear strength in one particular plane. Examples include graphite, molybdenium disulfide, and certain soaps.

Emulsion A liquid mixture in which an immiscible liquid is suspended in another liquid. Water and oil can be emulsified under certain conditions of oil type and severe agitation. Emulsifying agents are sometimes added to oils to produce cutting fluids that are to be mixed with water.

E.P. (extreme pressure) agent An additive introduced into a lubricant to impart load-carrying or antiweld qualities. E.P. agents are sometimes chemically active and produce what is called controlled chemical wear. Sulfur, chlorine, phosphorus, and lead soaps are often used for this purpose.

E.P. (extreme pressure) lubricant A lubricant that either has an E.P. agent added (see above), or that has inherent properties that permits it to carry appreciably higher loads than those carried by a non-E.P. lubricant.

EPA U.S. Environmental Protection Agency. Federal agency charged with administering and enforcing federal regulations governing air and water pollution control.

Ester A compound formed by the combination of an organic acid with an alcohol; e.g., acetic acid and ethyl alcohol give the ester ethyl acetate.

Evaporation loss The portion of a lubricant that volatilizes under the effects of temperature, pressure, and time. Such loss can occur in use or in storage. A widely used testing method is ASTM D-972-56.

Fat An animal or vegetable oil that will combine with an alkali to saponify and form a soap. (See Soap.)

Fatty acid The acidic part of a fat. Fatty acids are often used in the production of greases in lieu of whole fats. Such greases are more uniform, because the refining of the fat produces a more uniform raw material.

Feedability The ability of a lubricating grease to flow to the suction of a dispensing pump at a rate at least equal to pump delivery capacity. (Some lubricating greases do not feed satisfactorily and cause cavitation at the inlet to a dispensing pump. In such cases, feedability can often be made satisfactory by the use of a follower plate.) See also Dispensability.

Fiber In lubricating grease, the form in which soap thickeners occur, the soaps crystallizing in threads that are on the order of 20 or more times as long as they are thick. (Most soap fibers are microscopic in size, so that the grease appears smooth to the eye. The greases having a fibrous appear ance are those in which the fiber bundles are large enough to be seen by the naked eye. The most common fibrous lubricating grease is sodium base, but not all sodium-base greases are fibrous.)

Fibril An extremely small fiber, usually barely visible even at maximum magnification in an electron microscope. Fibrils may collect in bundles to form larger fibers.

Film strength The ability of a film of lubricant to resist rupture due to load, speed, and temperature.

Fire point The lowest temperature at which vapors released from an oil sur face will burn continuously for 5 sec minimum under specified conditions.

Flash point The lowest temperature at which a flash will occur over the sur face of an oil heated under specified conditions. This is usually about 50°F under the fire point.

Fluid-film lubrication That state of lubrication in which the shape and relative motion of the sliding surfaces causes the formation of a continuous fluid film under sufficient pressure to prevent any contact between the surfaces. Synonym: hydrodynamic lubrication.

Fluid friction A liquid's internal resistance to flow. (See also Viscosity.)

Follower plate A piece of sheet metal used on top of grease in a pumptype dis penser to ensure that the grease remains level as it is pumped. This avoids the formation of a cavity around the grease pump pick-up tube and permits the dispensing of all the grease in the container without manual leveling of the grease from time to time.

Fretting A form of wear resulting from oscillation or vibratory motion of lim ited amplitude that removes very finely divide particles from rubbing sur faces. In the case of ferrous metals in air, the wear particles oxidize to a reddish, abrasive iron oxide (Fe_2O_3), which has led to the condition being called fretting corrosion or, when it occurs in rolling contact bearings, false brinelling or friction oxidation.

Friction A resistance to motion between two surfaces in contact. (See also Coefficient of friction.)

Gallon (U.S.) Unit of liquid volume equal to 231 cu in., 3.785 liters, or 0.833 Canadian gallon. See Appendix D.

Gas The vapor state of any substance that has neither independent shape nor volume.

Gel An elastic solid mixture of a colloid and a liquid possessing a yield point and a jellylike texture.

Gram A metric unit of mass and weight equal to 1/1000 kg and nearly equal to the mass (weight) of 1 cm^3 of water at its maximum density. See Appendix D.

Gravity Given in two forms, either in degrees API or as specific gravity. The former is an arbitrary standard adopted by the American Petroleum Institute (the higher the number, the lighter the oil). Specific gravity is the ratio of the weight of oil to the weight of an equal volume of water.

Grease A lubricant composed of a lubricating fluid thickened with soap or other material to a solid or semisolid consistency.

Heat Energy released during combustion.

Heating value A measure of the amount of energy available in a fuel to produce work, expressed as Btu/lb or Btu/gal.

Homogenization The process of subjecting a lubricating grease to intimate mixing and intense shearing action, the end result of which is a more uniform and higher degree of dispersion.

Hydrated soap A soap that has water associated with its structure. A typical example is a water-stabilized calcium soap grease, which owes its stability to hydrated calcium soap.

Hydrocarbons Compounds containing only carbon and hydrogen. Petroleum consists chiefly of hydrocarbons.

Hydrodynamic lubrication See Fluid-film lubrication.

Hydrophilic Having an affinity for water; capable of uniting with or dissolving in water.

Hydrophobic Having an antagonism to water; not capable of uniting or mixing with water.

Hydrostatic lubrication That state of lubrication in which the lubricant is supplied to a plain bearing under sufficient external pressure to separate the opposing surfaces by a continuous lubricant film.

Hygroscopic Capable of readily absorbing water from the air.

Hypoid gear lubricant A gear lubricant having extreme pressure characteristics for use in hypoid-type gears such as automotive differentials.

Incompatibility A characteristic of two lubricants a mixture of which shows service performance physical properties that are markedly inferior to those of either lubricant before mixing. Performance or properties inferior to one of the products and superior to the other may be due to simple mixing and would not be considered as evidence of incompatibility.

Induction period (grease oxidation) The time during which oxidation occurs at a relatively low rate. The point at which the rate becomes significantly more rapid is the end of the induction period. There are several methods of measurement, including ASTM D-942-50.

Inhibitor A substance that slows or prevents chemical reactions such as oxidation or corrosion.

Inorganic thickener An inorganic material such as clay added to base oil to increase the viscosity or to thicken the material.

Insolubles Components of a lubricating grease that are insoluble in the prescribed reagents in an analytical procedure such as ASTM D-128-64. The analytical procedure used should be indicated when insolubles are specified. Additional identifying analytical tests are required to determine the nature and composition of insolubles, which may consist of fillers, additives, certain types of thickeners, or inadvertent impurities.

ISO International Standards Organization.

ISO viscosity A number indicating the nominal viscosity of an industrial fluid lubricant at 40°C (104°F) as denned by ASTM Standard Viscosity System for Industrial Fluid Lubricants D-2422-75. Essentially identical to ISO Standard 3448.

Kinematic viscosity Defined as absolute viscosity divided by density. Units are stokes (cm^2/sec). See Fig. C.2.

Lead scavengers Compounds of bromine and chlorine included in anti knock compounds to help remove lead compounds from the engine after combustion of fuel.

Lime-base grease See Calcium-base grease.

Lithium-base grease A grease prepared from a lubricating fluid thickened with lithium soap.

Load-carrying capacity Property of a lubricant to form a film on the lubri cated surface that resists rupture under given load conditions. Expressed as the maximum load the lubricated system can support with out failure or excessive wear. See also E.P agent; E.P. lubricant.

Lubricant Any substance interposed between two surfaces in relative motion that reduces the friction or wear of the surfaces.

Lubricating grease structure The physical arrangement of the component particles of a lubricating grease thickener, additive (if any), and liquid lubricant. It is the nature and stability of this arrangement that determines the appearance, texture, and physical properties of a lubricating grease.

Lubricity See Oiliness.

Luster The intensity of light reflected by lubricating grease, e.g., its sheen or brilliance. Luster can be described as follows:

Bright Reflects light with a relatively strong intensity.

Dull Reflects light with a relatively weak intensity. Some greases with a high water content may have a dull luster. Certain thickeners and fillers give a grease a characteristic dull luster.

Mechanical stability See Shear stability.

Metal deactivator An additive used in gasoline, diesel fuel, or lubricating

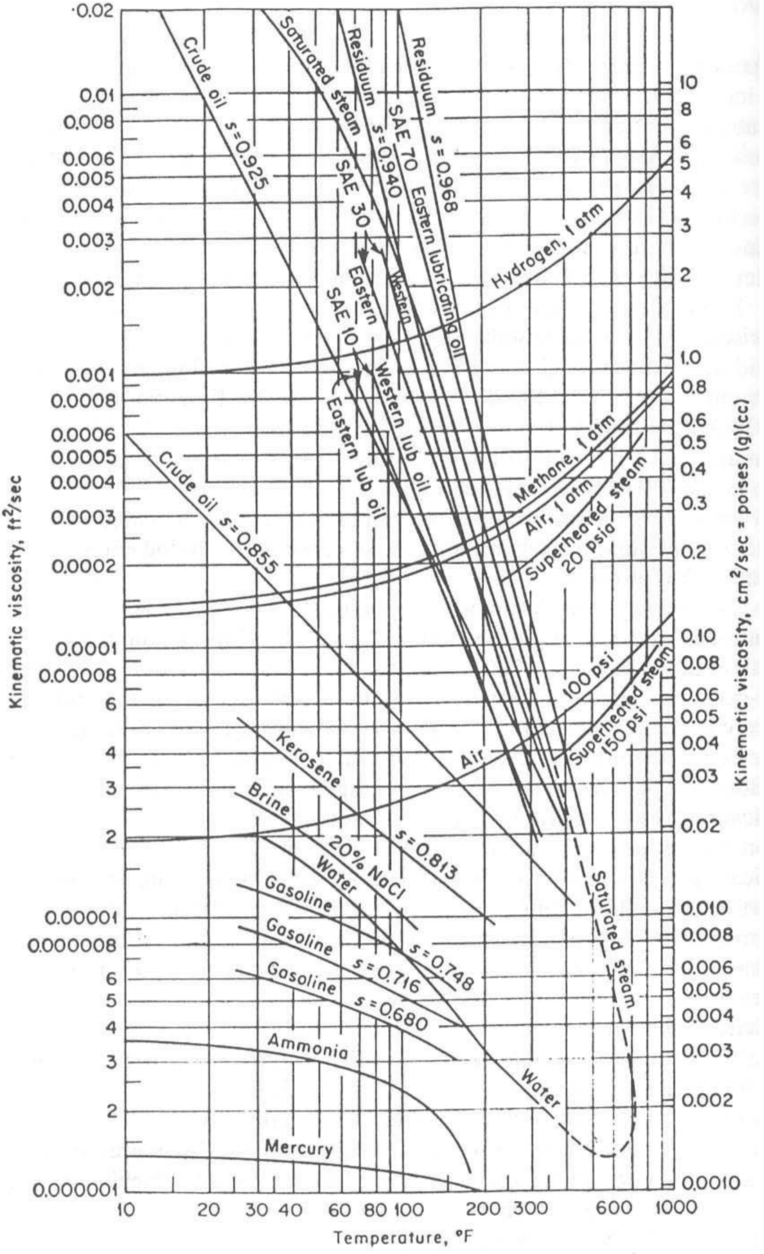


Fig. C.2 Kinematic viscosity versus temperature. [Courtesy of McGraw-Hill Fluid Mechanics Handbook (1968).]

oil to prevent the catalytic action of metal surfaces to promote oil oxidation and gum formation in gasoline and diesel fuel.

Mineral oil Term applied to a wide range of products with viscosities within the range of products considered petroleum-type oils.

Miscible Capable of forming a liquid solution or uniform mixture; e.g., gasoline and oil are miscible.

Mixed base grease A grease that is thickened with an agent composed of a mixture of more than one type of metallic soap.

Multigrade oil An oil meeting the viscosity limits of more than one SAE grade.

Neutral oil A low- or medium-viscosity oil obtained by distillation and treatment with solvent or clay. An oil not treated with acid or alkali.

Newtonian behavior A property of simple liquids in which the rate of shear is proportional to the shearing stress. This constant proportion is the viscosity of the liquid.

NLGI National Lubricating Grease Institute.

Non-Newtonian behavior The property possessed by some fluids and many plastic solids, including lubricating grease, of having a variable relationship between shear stress and rate of shear. Viscosity will vary with shear rate.

Non-soap thickener Any of several specially treated or synthetic materials, except the metallic soaps of long-chain fatty acids, that can be either thermally or mechanically dispersed in liquid lubricants to form lubricating grease. Also called synthetic thickener. Certain types are called inorganic thickeners.

Oil A water-insoluble viscous liquid that possesses lubricating properties.

Oiliness A term used to describe the ability of a lubricant to reduce friction between rubbing surfaces. There are no generally accepted test methods available to evaluate this property. Oiliness is important mostly in conditions of boundary lubrication and probably represents some relationship to the ability of the oil to wet the bearing surfaces and to resist being rubbed off. Oiliness has no known direct relationship to oil viscosity. Synonym: lubricity.

Organic material Material consisting essentially of carbon and hydrogen with or without small amounts of other chemical elements.

Oxidation The process of combining with oxygen. All petroleum products are subject to oxidation to some degree. The reaction increases with rise in temperature. With crankcase and other oils, oxidation produces oil-insoluble oxidized materials that increase viscosity and produce deposits.

Oxidation inhibitor An additive that minimizes the formation of harmful acids and varnish-forming compounds.

Oxidation stability The resistance of a lubricant to chemical reaction with oxygen. Absorption of and reaction with oxygen may lead to deterioration of the lubricant. Several testing methods are in use, including ASTM D-942-50 and Federal Test Method 2504.

Penetration An arbitrary measure of consistency (hardness), based on ASTM Method D-217-52T (and similar methods standardized by other organizations). (All penetration measurements are in an inverse scale of consistency; that is, the softer the consistency, the higher the penetration number.) The ASTM definitions are given as follows:

Penetration (of lubricating grease) The depth, in tenths of a millimeter, to which a standard cone penetrates the sample in a standard cup under prescribed conditions of weight, time, and temperature. The penetration depends on whether or not the consistency has been altered by handling or agitation.

Worked penetration The penetration of a sample of lubricating grease immediately after it has been brought to 77°F and then subjected to 60 double strokes in a standard grease worker.

Unworked penetration The penetration at 77°F of a sample of lubricating grease that has received only the minimum handling in transfer from sample can to test apparatus and that has not been subjected to the action of a grease worker.

Block penetration The penetration at 77°F of a sample of lubricating grease that is sufficiently hard to hold its shape.

Plasticity The property of an apparently solid material that enables it to be permanently deformed under the application of force, without rupture. (Plastic flow differs from fluid flow in that the shearing stress must exceed a yield point before any flow occurs).

Pour point The lowest temperature at which a liquid will pour under specified conditions.

Pour-point depressant A material that prevents insoluble wax molecules in oil from building a honeycomb (lattice-like structure) at colder temperatures. Particularly useful for paraffinic oils. Gives useful ability to pour at lower temperatures. An antifreeze for oil.

Pumpability The ability of a lubricating grease to flow under pressure through the line, nozzle, and fitting of a grease-dispensing system. Best indicated by the apparent viscosity at moderate rate of shear.

Residual fuel Fuel essentially composed of material remaining as unevaporated liquid after a distillation of crude oil. Usually high in sulfur but of relatively low cost. Widely used in high-output, slow-speed marine diesel engines.

Rheology Study of the deformation and/or flow of matter in terms of stress, strain, temperature, and time. The rheological properties of lubricating greases are commonly measured by penetration and apparent viscosity.

Rheopectic grease A grease that will become harder (more consistent) when it is worked or agitated.

SAE Society of Automotive Engineers.

SAE number (grade) Number applied to crankcase, transmission, or rear axle lubricant to indicate its viscosity range.

Saponification The process of forming soap. See also Soap.

Saponification number The number of milligrams of potassium hydroxide required to react with all fats and fatty acids in a 1 g sample. A measure of the degree of compounding in a gear oil, stream cylinder oil, or other compounded lubricant.

Set In the manufacture of a lubricating grease, the change from a fluid to a semifluid or plastic state.

Sett grease A lubricating grease made by mixing a mineral oil slurry of an alkali and a mineral oil solution of a fatty acid. Saponification is usually completed after the mixture is poured into the shipping container. As the temperature of the mixture is generally less than 140°F, such a product is sometimes known as a cold setts. Axle grease is usually a sett grease.

Shearing Slipping or sliding of one part of a substance relative to an adjacent part. In a solid, such action involves cutting or breaking of the crystal structure, but in a fluid or plastic, shearing does not necessarily destroy the continuous nature of the substance.

Shear rate The rate of slip within a substance engaging in flow. The average or mean shear rate in a pipe or tube is the average velocity divided by the radius of the tube. It therefore has the dimensions of the reciprocal of time and is usually expressed in units of reciprocal seconds (sec-1). The mean shear rate is reported in the determination of apparent viscosity in ASTM Method D-1092-55.

Shear stability The ability of a lubricating grease to resist changes in consistency (hardness) during mechanical working. Working can be in any of several types of laboratory machines or in actual service. Also called mechanical stability.

Shear stress The force required to cause shearing in a substance. In fluids, the relation of the shear stress to the shear rate is the viscosity of the substance.

Slumpability The property of some greases that makes them partially self-leveling. Slumpable greases can be pumped from a container without having to use a Follower plate. Often confused with pumpability, which see.

Soap A specific type of salt formed by the reaction of a fatty acid with an alkali.

Sodium-base grease A grease prepared from a lubricating fluid thickened with sodium soap. Also called soda grease.

Sol A suspension of particles of colloidal dimensions in a liquid that possesses the gross properties of a liquid.

Solid lubricant Any of a class of lubricants in which the reduction of friction and wear during sliding is caused by making the shearing take place within the crystal structure of a material with low shear strength in one particular plane. Examples include graphite, molybdenum disulfide, and certain soaps. Lubricating grease is not a solid lubricant but may contain solid lubricants as additives.

Specific gravity The ratio of the weight (in air) of a given volume of a material to the weight (in air) of an equal volume of water at a stated temperature.

Squeeze-film lubricant A lubricant that induces that state of lubrication in which surfaces thickly coated or flooded with lubricant move toward each other at sufficient speed to develop fluid pressure sufficient to support a load of short duration. Because of viscosity (or apparent viscosity), the lubricant cannot immediately flow away from the area of contact. This action occurs, for example, between gear teeth and between wrist pins and their bushings.

SSU Saybolt seconds universal. A unit of measure of lubricating oil viscosity commonly used in the United States.

Straight mineral oil A petroleum oil not containing compounds (animal or vegetable oils) or chemical additives.

Syneresis Loss of liquid lubricant from a lubricating grease due to shrinkage or rearrangement of the structure. The shrinkage may be due to either physical or chemical changes in the thickener. Syneresis is a form of bleeding.

Synthetic oil An oil produced by synthesis rather than by extraction or refinement.

Synthetic thickener See Non-soap thickener.

Tacky A descriptive term applied to a grease that appears to be particularly sticky or adhesive.

TBN Total base number. A measure of the ability of a lubricant to neutralize acids. High TBN is desirable in a crankcase oil to control corrosive engine wear from acids of combustion.

Texture The property of a lubricating grease that is observed when a small separate portion of it is pressed together and then slowly drawn apart. Texture can be described in the following terms:

Brittle Has a tendency to rupture or crumble when compressed.

Buttery Separates in short peaks with no visible fibers.

Long fiber Shows tendency to stretch or string out into a single bundle of fibers.

Short fiber Shows short break-off with evidence of fibers.

Resilient Capable of withstanding moderate compression without permanent deformation or rupture.

Stringy Shows tendency to stretch or string out into long fine threads but with no visible evidence of fiber structure.

Thermal stability Ability to resist oxidation under high-temperature operating conditions.

Thickening agent A solid whose particles can be relatively uniformly dispersed to form the structure of lubricating grease in which the liquid is held by surface tension and other physical forces. The solid particles may be fibers as in various metallic soaps, or plates of spheres, as in some non-soap thickeners. The only general requirements are that the particles be extremely small and capable of uniform dispersion in the liquid lubricant.

Thixotropy In lubricating grease, the property manifested by a decrease in consistency or softening as a result of shearing, followed by an increase in consistency or hardening, beginning after shearing is stopped. Thixotropic age hardening is a relatively prolonged process proportional to aging time and is seldom, if ever, complete, whereas the apparent viscosity increase that occurs in non-Newtonian systems with decreasing shear rate is instantaneous and fully reversible. Lubricating grease is both thixotropic and non-Newtonian.

Twelve-hydroxy 12-Hydroxystearate, a stearate soap made with some metal combined with fatty acid prepared from castor bean oil that has been hydrogenated. 12-Hydroxystearate soaps produce greases that are unusually stable to shear.

Uncompounded oil Straight mineral oil.

Unsaturated A term applied to carbon compounds in which some of the carbon atoms are held together by double bonds, so that these compounds are capable of combining directly with other elements or compounds. Lubricants containing unsaturated materials are usually somewhat unstable and subject to oxidation.

V.I. improver A material added to a fluid that alters the viscosity characteristic of the fluid and increases its viscosity index (V.I.).

Viscosity The internal friction of a fluid or semifluid; its resistance to flow.

Viscosity index (V.I.) A term for expressing the degree of change in a fluid's viscosity relative to a change in temperature. High degrees of viscosity change with temperature variations indicate low viscosity indices, while a low degree of viscosity change is characteristic of high viscosity index material. A high viscosity index is generally considered a point of quality in lubricants.

Volatility The ease with which a liquid is converted into a vapor state. High volatility is desirable in fuels. Low volatility is generally desirable in lubricating oils.

Water-corrosion resistance The ability of a lubricating grease to prevent corrosion of surfaces when water is present in the lubricating system.

Water absorption characteristics Amount of water a grease will absorb or retain.

Water resistance The ability of a lubricating grease to resist being removed from a bearing when acted on by a stream of water.

Wear The removal of materials from surfaces in relative motion.

Worked penetration The penetration of a sample immediately after it has been brought to a temperature of 77°F and then subjected to 60 double strokes in a standard grease worker. See also Penetration.

Working Subjecting lubricating grease to any form of agitation or shearing action.

Yield (of lubricating grease) The amount of grease of a given consistency that can be made with a specified amount of thickening agent. As the yield increases, percentage of thickener decreases.

Yield point The minimum force required to produce flow of a plastic material. It is estimated by the interception on the shear stress axis of the shear stress--shear rate curve, by extrapolation of the straight portion of the curve. Synonym: Yield value

Yield value See Yield point.

APPENDIX D

System of Units and Conversion Tables

A comparison of base and derived units for the SI, British, and metric (cgs, mks) systems appears in Table D.1.

Conversion from one set of units to another for various parameters are given in Tables D.2--D.9.

Table D.2 Absolute Viscosity
Table D.3 Flow Rate
Table D.4 Force
Table D.5 Mass
Table D.6 Pressure
Table D.7 Volume
Table D.8 Work, Energy, Heat
Table D.9 Work Rate (Power)

To use these tables, first find the column containing the parameter to be converted. Then find the row in that column that contains unity (1). The numbers in that row are the multipliers to use to convert the given unit to the other units.

Example. Convert 3 kilowatts into other units of power. The conversions for power are found in Table D.9. In the column under kilowatts, find the row containing unity (1). The numbers in that row must be multiplied by the number of kilowatts (in this case 3) to effect the conversion.

$$3 \text{ kilowatts} = (3 \text{ kW})\left(1.34 \frac{\text{hp}}{\text{kW}}\right) = 4.02 \text{ hp}$$

$$3 \text{ kilowatts} = (3 \text{ kW})\left(737 \frac{\text{ft-lb}_f/\text{sec}}{\text{kW}}\right) = 2211 \frac{\text{ft-lb}_f}{\text{sec}}$$

$$3 \text{ kilowatts} = (3 \text{ kW})\left(\frac{1000 \text{ W}}{\text{kW}}\right) = 3000 \text{ W}$$

$$3 \text{ kilowatts} = (3 \text{ kW})\left(102 \frac{(\text{kg}_f\text{-m/sec})}{\text{kW}}\right) = 306 \frac{\text{kg}_f\text{-m}}{\text{sec}}$$

$$3 \text{ kilowatts} = (3 \text{ kW})\left(0.238 \frac{\text{kcal/sec}}{\text{kW}}\right) = 0.714 \text{ kcal/sec}$$

$$3 \text{ kilowatts} = 3 \text{ kW}\left(1.36 \frac{\text{PS}}{\text{kW}}\right) = 4.08 \text{ PS}$$

Table D.1 A Comparison of Units

Quantity	Symbol	SI system	British system	Metric system
Base Units				
Length	L,l	meter (m) 1 m = 39.37 in	inch (in.) 1 in. = 0.0254 m	centimeter (cm) 1 cm = 10-2 m
Mass	m	kilogram (kg) 1 kg = 2.205 lb	pound mass (lb) 1 lb = 0.4535 kg	gram (g) or kilogram (kg)
Time	t	second (s)	second (s)	second (s)
Electric current	l	ampere (A)	ampere (A)	ampere(A)
Temperature	T	kelvin (K) T(K) = T(°C) + 273.15	degree Fahrenheit (°F) or degree Rankine (°R) T(°F) = 1.8T(°C) + 32 T(°R) = T(°F) + 459.67	degree Celsius (°C) T(°C) = [T(°F) - 32]/1.8
Derived Units				
Acceleration	a	meter per second squared (m/s^2)	foot per second squared (ft/s^2)	centimeter per second squared (cm/s^2)
Area	A	square meter (m^2)	square inch ($in.^2$)	square centimeter (cm^2)
Density	ρ	kilogram per cubic meter (kg/m3)	pound mass per cubic inch (lb/in.3)	grams per cubic centimeter (g/cm3)
Energy or work	E or W	joule (J) 1 J = 1 N-M	foot pound force (ft-lbf) 1ft-lbf = 1.356 J	erg (= 1 dyn-cm) 1 erg = 10-7 J
Flow rate (volumetric)	Q,q	cubic meter per second (m^3/s)	cubic inch per second ($in.^3/s$) gallons per minute (gpm) 1gpm = 3.7854 L/m	cubic centimeter per second (cm^3/sec) liter per minute L/min
Force or weight	F,f	newton (N) 1 N = 1 kg-m/s^2	pound force (lb_f) 1 lb_f = 4.448 N	kilogram force kg_f 1 kg_f = 9.81 N
Frequency	f	hertz (Hz) 1 Hz = 1 cycle/s	hertz (Hz)	hertz (Hz)

Table D.1 Continued

Quantity	Symbol	SI system	British system	Metric system
Power	P	watt (W) 1 W = 1 J/s	foot pound force per second (ft-lb$_f$/s) 1 ft-lbf/s = 1.356 W horsepower (hp) 1 hp = 746 W	metric horse power (PS) 1 PS = 735 W
Pressure	p,P	Pascal (pa) 1 Pa = 1 N/m^2 = 10^{-5} bar	pound force per square inch (psi) lb$_f$/in^2 1 psi = 6897 Pa	kilogram force per square centimeter (kg$_f$/cm^2) 1 kg$_f$/cm^2 = 98067 Pa
Velocity	v	meter per second (m/s)	foot per second (ft/s)	centimeter per second (cm/s)
Volume	V	cubic meter (m^3)	cubic inch (in.^{3}) gallon (gal) 1 gal = 231 in.^{3} = 3.7854 L	cubic centimeter (cm^3) liter 1 L = 10^{3}cm^3 = 10^{-3} m^3

Table D.2 Absolute Viscosity Conversion Factors

Absolute Viscosity = Kinematic Viscosity x Fluid Density

centipoise (cp)	poise (p)	Pa-s (N-s/m^2)	kg_f-s/m^2	reyns (lb_f-s/$in.^2$)	lb_f-s/ft^2
1	0.01	0.001	10.2 x 10^{-5}	1.4503 x 10^{-7}	2.0886 x 10^{-5}
100	1	0.1	10.2 x10^{-3}	1.4503 x 10^{-5}	2.0886 x 10^{-3}
1000	10	1	0.102	1.4503 x 10^{-4}	2.0886 x10^{-2}
9810	98.1	9.81	1	1.422 x 10^{-3}	2.0476 x 10^{-1}
6.895 x 10^{6}	6.895 x 10^{4}	6.895 x 10^{3}	7.033 x 10^{2}	1	6.9441 x 10^{-3}
9.9288 x 10^{8}	9.9288 x 10^{6}	9.9288 x 10^{5}	1.0128 x 10^{7}	144.01	1

Table D.3 Flow Rate Conversion Factors

gpm (U.S.) (gal/min)	gpm (U.K.) (gal/min)	cis (in.^{3}/s)	lpm (liter/min)	lps (liter/s)	ccps (cm^3/s)
1	0.8325	3.85	3.7854	6.304 x 10^{-2}	63.04
1.2011	1	4.621	4.5465	7.5775 x 10^{-2}	75.775
0.25974	0.21622	1	0.98312	1.63862 x 10^{-2}	16.3862
0.26419	0.21995	1.0172	1	0.0167	16.7
15.8514	13.197	61.0304	60	1	10^{+3}
0.015851	0.013197	0.006103	0.06	10^{-3}	1

Table D.4 Force Conversion Factors

pound force (lb_f)	kilogram force (kg_f)	newton (N)	dyne (dyn)[a]
1	0.45359	4.44792	444792
2.20462	1	9.80665	980665
0.2248	0.102	1	10^5
2.248 x 10^{-6}	1.0197 x 10^{-6}	10^{-5}	1

[a] 1 dyn = 1 g-cm/s^2.

Table D.5 Mass Conversion Factors

slug	short tons	long tons	ounces	pounds	grains	kg
1	1.6083 x 10^{-2}	1.4364 x 10^{-2}	514.7402	32.1804	225218.91	14.5943
62.1779	1	0.89310	32005.4	2000.907	14003629.8	907.44102
69.61999	1.1196912	1	35836.21215	2240.3983	15679740	1016.05365
1.94273 x 10^{-3}	3.12447 x 10^{-5}	2.79047 x 10^{-5}	1	6.2518 x 10^{-2}	437.53985	2.8353 x 10^{-}
3.1075 x 10^{-2}	4.9973 x 10^{-4}	4.46349 x 10^{-4}	15.99546	1	6998.63946	0.453515
4.4401 x 10^{-6}	7.1410 x 10^{-8}	6.37766 x 10^{-8}	2.2855 x 10^{-3}	1.42885 x 10^{-4}	1	6.48 x 10^{-5}
6.852 x 10^{-2}	1.102 x 10^{-3}	9.842 x 10^{-4}	35.27	2.205	15432	1

Table D.6 Pressure Conversion Factors

Pounds per sq in. (lbf/in.^{2})	Atmosphere (atm)	Bars (10^5 N/m^2)	Pascals (N/m^2)	kg$_f$/cm^2	Millimeters of mercury (mm Hg)
1	6.897 x 10^{-2}	6.897 x 10^{-2}	6.897 x 10^{3}	7.3233 x 10^{-2}	51.7408
14.70	1	1.013	101325	1.0330	760
14.50	0.9869	1	105	1.02	750.1
14.5 x 10^{-5}	9.869 x 10^{-6}	10^{-5}	1	1.02 x 10^{-5}	0.0075
14.22	0.9678	0.98066	98067	1	735.6
1.934 x 10^{-2}	0.00132	1.33 x 10^{-3}	133.32	0.00136	1

Table D.7 Volume Conversion Factors

Cubic inch (in.^{3})	Cubic foot (ft^3)	Gallon (gal, U.S.)	Gallon (gal, U.K.)	Milliliter[a] (mL)	Liter (L)	Cubic meter (m^3)
1	0.00058	0.004329	0.0036	16.39	0.01639	0.00001639
1728	1	7.4805	6.2305	28317	28.317	0.02832
231	0.1338	1	0.8331	3785.4	3.7854	0.00379
277.27	0.1605	1.2003	1	4543.7	4.5437	0.00454
6.103 x 10^{-2}	3.53 x 10^{-5}	2.642 x 10^{-4}	2.2 x 10^{-4}	1	0.001	1 x 10^{-6}
61.03	0.0353	0.2642	0.22	1000	1	0.001
61024	35.315	264.2	219.975	1 x 10^{6}	1000	1

[a] 1 mL = 1 cm^3.

Table D.8 Work, Energy, Heat Conversion Factors

Horsepower-hours (hp-br)	British thermal units (Btu)	Foot-pounds$_f$ (lbf-ft	Kilowatt hours (kWh)	Joules[a] (J)	Kilogram$_f$-meters /s (kg$_f$-m)
1	2.544 x 10^{3}	1.981 x 10	0.7457	2.686 x 10^{6}	2.739 x 10^{5}
3.931 x 10^{-4}	1	7.783 x 10^{2}	2.931 x 10^{-4}	1.055 x 10^{3}	107.63
5.049 x 10-7	1.285 x 10^{-3}	1	3.766 x 10^{-7}	1.356	0.1383
1.3405	3.412 x 10^{3}	2.655 x 10^{6}	1	3.6 x 106	3.671 x 10^{5}
3.724 x 10^{-7}	9.477 x 10^{-4}	0.7376	2.778 x 10^{-7}	1	0.102
3.3652 x 10-6	9.291 x 10^{-3}	7.233	2.724 x 10^{-6}	9.807	1

[a] 1 J = 1 N-m.

Table D.9 Work Rate (Power) Conversion Factors

Horsepower (hp)	Foot - pound/s (ft-lb$_f$/s)	Watt[a] (W)	Kilowatt (kW)	Kilogram$_f$-meter/s (kg$_f$/s)	Kilocalorie/s) (kcal/s	Metric horsepower (PS)
1	550	746	0.746	76	0.178	1.014
0.00182	1	1.356	0.00136	0.138	0.000324	0.00184
0.00134	0.737	1	0.001	0.102	0.000238	0.00136
1.34	737	1000	1	102	0.238	1.36
0.0131	7.233	9.807	0.00981	1	0.00234	0.01333
5.61	3090	4200	4.2	427	1	5.7
0.987	541	735	0.735	75	0.175	1

[a] 1 W = 1 N-m/s.

APPENDIX E

Nomenclature

α	Life constant for a given material	(hr)
α	Air-side angle	(degrees)
α	The angle from the center of the shaft split to any point on the shaft surface	(degrees)
$d\alpha$	Differential angle	(degrees)
ΔT	Time increment	(hr)
θ	Characteristic life	(hr)
θ	Oil-side angle	(degrees)
γ	Point on seal contact band	(degrees)
γ_{in}	Point on seal contact band where in-pumping stops	(degrees)
γ_{out}	Point on seal contact band where out-pumping begins	(degrees)
π	3.14159	
μ	Coefficient of friction	(dimensionless)
A	Cross-sectional area of lip	(mm^2; in.^{2})
AID	Spring assembled inner diameter	(mm; in.)
A_{in}	Dimensionless parameter	
A_{out}	Dimensionless parameter	
Am	Amplitude of wave on seal lip	(mm; in.)

B	Weibull curve shape parameter (dimensionless)	
C	Elastomer compression factor (about 0.85)	
C	Dimensionless parameter	
C_p	Process capability, dimensionless	
C_{pk}	Modified process capability, dimensionless	
D_B	Bore diameter	(mm; in)
D_C	Diameter to center of gravity of lip cross section	(mm; in.)
D_L	Primary lip inner diameter	(mm; in.)
D_O	Seal outer diameter	(mm; in.)
D_s	Shaft diameter	(mm; in.)
D_1	Diameter to center of flex point ($=D_2 + t$)	(mm; in.)
D_2	Heel diameter (=DL + 2h)	(mm; in.)
E	Energy of activation	(dimensionless)
E_A	Modulus of elastomer at 5% strain (against grain)	(N/mm^2; psi)
E_W	Modulus of elastomer at 5% strain (with grain)	(N/mm^2; psi)
F	Spring tension	(N; lb)
F	Incremental force exerted by seal lip	(N; lb)
F	Failure rate	(hr)
F_B	Force due to rubber beam deflection	(N; lb)
F_i	Force sensed by instrument	(N; lb)
F_R	Force due to rubber hoop forces	(N; lb)
F_S	Force due to spring	(N; lb)
F_T	Total force	(N; lb)
F_T	Cumulative failure (%)	
F_{T50}	Cumulative percent failure for median rank (%)	
G	Duty parameter (dimensionless)	
h	Film thickness	(mm; in.)
H	Distance from contact point to heal	(mm; in.)
I	Interference between shaft and seal lip	(mm; in.)
j	Rank of functional test for Weibull analysis (dimensionless)	
L	Seal life	(hr)
L	Distance from flex point to lip contact point	(mm; in.)
L_c	Distance from flex point to center of gravity of lip section	(mm; in.)
Le	Leakage rate of seal without hydrodynamic aids	(cm^3/min)

Le_B	Leakage rate of seal installed backward without hydrodynamic aids	(cm^3/min)
Leh	Leakage rate of seal with hydrodynamic aids	(cm^3/min)
Leh_B	Leakage rate of seal installed backward with hydrodynamic aids	(cm^3/min)
L_{NBR}	Seal life with NBR compound	(hr)
L_{PA}	Seal life with polyacrylate compound	(hr)
L_{SL}	Lower specification limit	(units as required)
M_{HF}	Maximum elastomer torque measured with oscillating disk rheometer	(N-m; lb-ft)
M_L	Minimum elastomer torque measured with oscillating disk rheometer	(N-m; lb-ft)
MV	Minimum value of elastomer viscosity measured with Mooney machine	(N-m; lb-ft)
n	Number of waves on seal lip (dimensionless)	
N	Shaft speed	(rev/s)
N	Number of measurements of seal characteristic (dimensionless)	
N_F	Number of failures during time frame ΔT (dimensionless)	
N_{FC}	Cumulative number of failures (dimensionless)	
N_T	Total number of seals tested (dimensionless)	
P	Total force between seal and shaft	(N;lb)
Pf	Seal O.D. press fit	(mm; in.)
P_{hp}	Power	(hp)
P_{kW}	Power	(kW)
PT	Pump time	(s)
Q	Pump rate	(cm^3/s)
Q*	Dimensionless pump rate	
Q_{net}	Net volumetric flow into/out of the sump	(cm^3/s)
$Q_{net\infty}$	Net volumetric flow at infinite shaft speed	(cm^3/s)
Q_{norm}	Normalized pump rate (= $2\pi R_s hVs\mu$.)	(cm^3/s)
R	Universal gas constant (dimensionless)	
R_a	Roughness average	(mm; in.)
R_C	Rockwell hardness "C" scale (dimensionless)	
R_{CS}	Radial cross section (dimensionless)	
Re	Reliability %	
R_L	Radial load	(N/mm; lb/in.)
R_r	R value	(mm; in.)
R_s	Shaft radius	(mm; in.)
RWD	Radial wall dimension	(mm; in.)

RWV	Radial wall variation	(mm; in.)
S	Standard deviation of sample	(various units of measurement)
Sc	Sealing capacity	(dimensionless)
Sc_B	Sealing capacity using backward installation	(dimensionless)
t	Flex or beam thickness	(mm; in.)
t_a	Thickness of elastomer pill after compression set test	(mm; in.)
t_b	Thickness of elastomer pill before test	(mm; in.)
T_C	Time for elastomer to reach a 90% curve level as measured with oscillating disk rheometer	(min)
T_{s2}	Scorch time of elastomer measured with oscillating disk rheometer	(min)
T	Total radial force of seal	(N; lb)
T	Observed life of seal	(hr)
T_K	Sump temperature	(K)
T_0	Weibull curve location parameter	(hr)
TIR	Total indicated reading	
Tor	Torque	(N-m)
Tor*	Torque	(in.-oz)
T_1	Time for first failure	(hr)
T_2	Time for midpoint failure from Weibull curve	(hr)
T_3	Time of last failure	(hr)
T_5	Time for Mooney reading to increase by 5 units	(min)
T_{35}	Time for Mooney reading to increase by 35 units	(min)
$T\Delta_{30}$	Difference between T35 and T5	(min)
USL	Upper specification limit	(units as required)
V_{in}	Inward velocity	(mm/s; in./s)
$V_{oi}l$	Oil droplet volume	(cm3)
V_{out}	Outward velocity	(mm/s; in./s)
V_s	Linear shaft velocity	(mm/s; in./s)
W	Seal contact width	(mm; in.)
W_b	Bore width	(mm; in.)
W_d	Wire diameter	(mm; in.)
W_s	Seal width	(mm; in.)
$\overline{X}$	Average value of the seal characteristic	(units as required)
X_i	Specific value of the seal characteristic	(units as required)
Z_{cp}	Fluid viscosity	(N-s/mm^2)